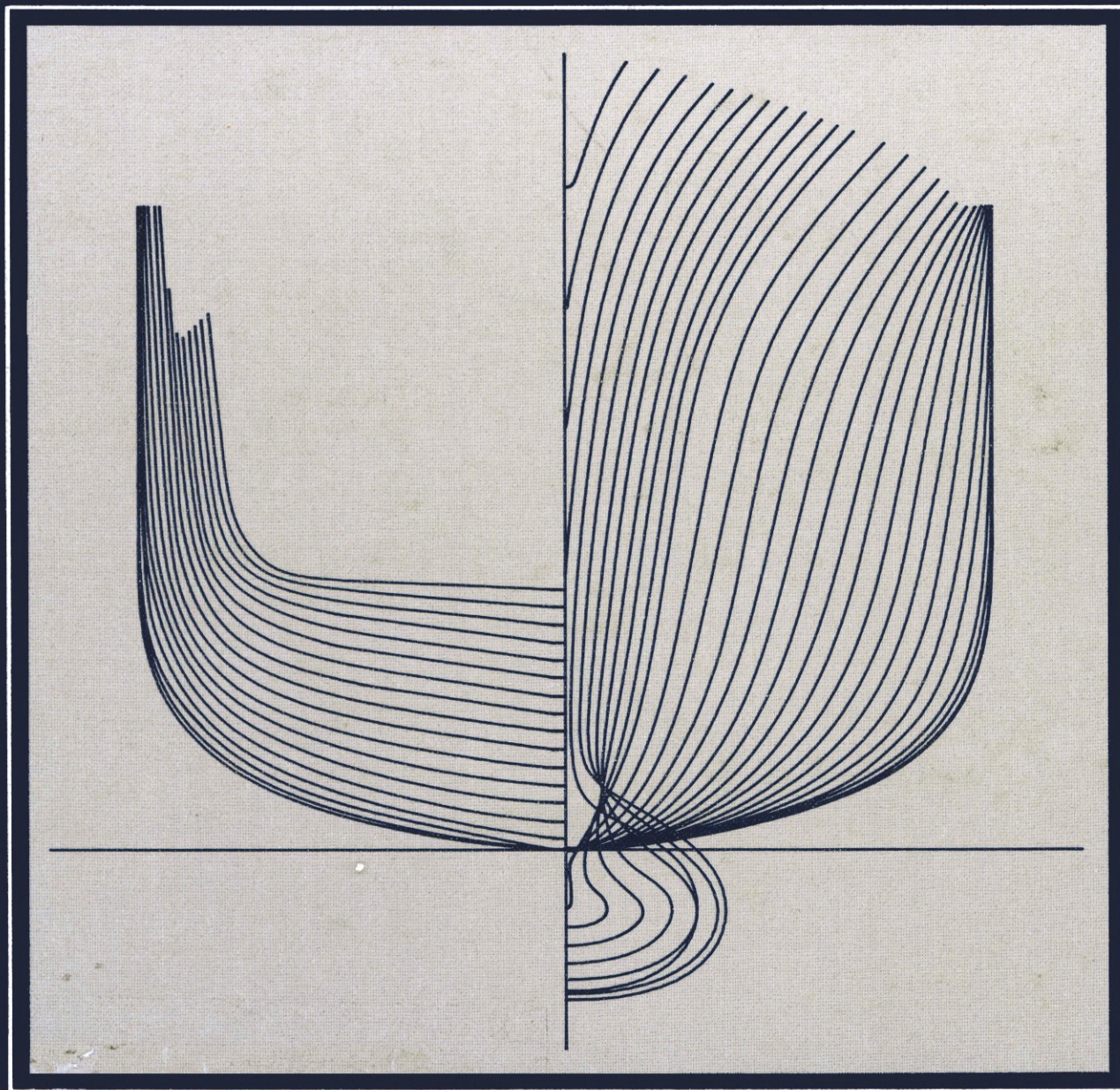


Introduction to **NAVAL ARCHITECTURE**



By Thomas C. Gillmer and Bruce Johnson

Introduction to Naval Architecture

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BY THOMAS C. GILLMER AND BRUCE JOHNSON



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Preface

This textbook has been prepared to satisfy the educational requirements of the Naval Systems Engineering Department at the U.S. Naval Academy. The department offers two third-class (sophomore-level) courses that teach engineering fundamentals of naval architecture, especially those connected with naval ship design. A four-semester-hour course, Naval Engineering I, is taught to most of the non-engineering majors at the Naval Academy whose background includes chemistry, physics, and mathematics through differential equations. Because these students have not had the engineering fundamentals courses, this textbook contains a brief summary of engineering statics (chapter 2), engineering materials (chapter 4), strength of materials (chapter 5), and fluid mechanics (chapter 11).

A three-semester-hour course, Introduction to Naval Systems Engineering, is taught to majors in ocean engineering and marine engineering to give them a background in naval architecture and to acquaint them with aspects of engineering design early in the curriculum. It is hoped that this text will satisfy the requirements of other institutions that cover this subject matter at the sophomore level. Engineering majors can skip sections of chapters 2, 4, and 5, if they have already had courses covering this material, in order to complete most of the book during a three-semester-hour course.

The text does not attempt to exhaust the subject of naval architecture. The general design of merchant ships, cargo-handling equipment, and habitability systems, as well as costing and contracting, computer-aided ship design, ship construction, launching, trials preparation, and delivery are omitted from the text. These subjects are extensively covered in a SNAME publication, *Ship Design and Construction*, to which this book frequently refers.

The text makes extensive use of material from *Modern Ship Design* by Professor Thomas Gillmer. Professor Bruce Johnson has contributed a considerable amount of new material, including example problems.

Because of the desired 1982 publication date, there was insufficient time to convert existing material to dual English-metric units. This will be accomplished for the second edition of the book. The symbols generally conform to those adopted by the International Towing Tank Conference (ITTC) and the International Ship Structures Conference (ISSC), except in those cases where symbols have not yet been standardized or where certain symbols have been chosen for pedagogical reasons. Professor Johnson is currently responsible for updating the international standard symbol list as a member of the Information Committee for the Nineteenth ITTC, which will meet in 1984.

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Critical reviews of portions of the manuscript were supplied by Dr. John Ince (chapters 1–10), Dr. Robert Pond (chapter 4), Mr. Robert Peach (chapters 4 and

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The authors acknowledge the support and encouragement of the Naval Institute staff, especially Mr. Richard Hobbs, who guided the project, Ms. Constance Buchanan, who carefully edited the manuscript, and Ms. Beverly Baum and Ms. Cynthia Taylor, who coordinated the layout and artwork. The authors also acknowledge the assistance of Mrs. Dorothy Johnson throughout the project.

Much of the basic material in this text is borrowed from the source book *Modern Ship Design* by Thomas C. Gillmer. The authors are indebted to those who cooperated with him in the two editions of that volume, which is still in print today.

Symbols and Abbreviations

The following symbols are considered the most applicable and convenient notations for use by a student of naval architecture. In most cases, they conform to the standard usages approved by the Society of Naval Architects and Marine Engineers and the International Towing Tank Conferences. In certain instances, minor deviations to avoid duplication are considered more advantageous for use in this text. Numerical subscripts following a symbol denote successive locations, progressions, or values of the basic symbol.

A —area, in general	b —span of a control surface (perpendicular to direction of flow)
A_M —area, midships section	$\bar{b}$ —mean span of a control surface
A_O —area of propeller disk	C_A —model-ship correlation allowance
A_P —projected area of a foil or propeller disk	C_B —block coefficient, ∇/LBT
A_W —area, waterplane	C_D —drag coefficient, $D/^{1/2}\rho Av^2$
A_X —area, maximum transverse section	C_F —frictional-resistance coefficient, $R_F/^{1/2}\rho Sv^2$
AP —after perpendicular	C_{IL} —longitudinal-waterplane inertia coefficient, $12I_L/L^3B$
AR —aspect ratio, $\bar{b}/\bar{c}$	C_{IT} —transverse-waterplane inertia coefficient, $12I_T/LB^3$
$\overline{AF}$ —distance from after perpendicular to center of flotation	C_L —lift coefficient, $L/^{1/2}\rho Av^2$
$\overline{AG}$ —distance from assumed center of gravity to actual center of gravity	C_M —midship section coefficient, A_M/BT
$\overline{AZ}$ —horizontal distance from assumed center of gravity to Z	C_P —prismatic coefficient, ∇/A_XL
a —area (generally a small area within a system)	C_R —residual-resistance coefficient, $R_R/^{1/2}\rho Sv^2$
a —linear acceleration	C_S —wetted-surface coefficient, $S/\pi BL$
B —beam or breadth, molded, of a ship	C_T —total-resistance coefficient, $R_T/^{1/2}\rho Sv^2$
B —position of center of buoyancy	C_V —viscous-resistance coefficient, $R_V/^{1/2}\rho Sv^2$
B_1 (etc.) —changed position of the center of buoyancy	C_{VP} —vertical prismatic coefficient, ∇/A_WT
BHP —brake horsepower	C_W —wavemaking-resistance coefficient, $R_W/^{1/2}\rho Sv^2$
$\overline{BM}$ —transverse metacenter above center of buoyancy (metacentric radius)	C_{WP} —waterplane-area coefficient, A_W/LB
$\overline{BM}_L$ —longitudinal metacenter above center of buoyancy (longitudinal metacentric radius)	C_X —maximum-transverse-section coefficient, $A_X/(BT)_X$
b —width of a compartment or tank	c —chord of a control surface (parallel to direction of flow)
b —center of buoyancy of an added buoyant layer	$\bar{c}$ —mean chord of a control surface
	c —damping coefficient
	c —distance from neutral axis to extreme fiber
	c_w —celerity or phase velocity of a wave
	$CL(\text{C})$ —centerline
	$\overline{D}$ —drag force
	D —diameter, generally
	D —molded depth of a ship hull
	DWL —designed load waterline
	DWT —deadweight tons
	dx —increment of length along an axis (also dy , dz , dA , etc.)

- E —modulus of elasticity or Young's modulus
 E —energy, generally
 EHP —effective horsepower
 e —elongation
 F —position of center of flotation (centroid of waterplane)
 F_1 —changed position of the center of flotation
 F —force, generally
 F_B —buoyancy force
 F_n —Froude number, $v/\sqrt{gL}$
 F_R —resultant force
 $\overline{FB}$ —distance from forward perpendicular to center of buoyancy
 $\overline{FF}$ —distance from forward perpendicular to center of flotation
 $\overline{FG}$ —distance from forward perpendicular to center of gravity
 FP —forward perpendicular
 FW —fresh water (usually taken as 36 cu ft per ton)
 f —coefficient of friction
 f —frequency, cycles per unit time
 f_E —frequency of encounter
 G —position of ship's center of gravity
 G_1 (etc.) —changed position of ship's center of gravity
 G —shear modulus, $E/2(1 + \mu)$
 $\overline{GG}_1$ —distance through which the ship's center of gravity moves
 $\overline{GM}$ —transverse metacentric height, height of M above G
 $\overline{GM}_L$ —longitudinal metacentric height, height of M_L above G
 $\overline{GZ}$ —righting arm; horizontal distance from G to Z
 g —acceleration of gravity
 g —position of center of gravity of a component weight
 g_1 —changed position of center of gravity of a component weight
 H —wave height, trough to crest
 h —depth of water or of submergence
 I —moment of inertia, generally
 I_L —longitudinal moment of inertia of waterplane
 I_T —transverse moment of inertia of waterplane
 I_x —moment of inertia of hull girder section about neutral axis
 i —moment of inertia of free surface in a compartment
 i —incrementing integer in a summation
 J —advance coefficient, v_A/nD
 K —any point in a horizontal plane through the baseline (keel)
 K_Q —torque coefficient, $Q/\rho n^2 D^5$
 K_T —thrust coefficient, $T/\rho n^2 D^4$
 $\overline{KB}$ —distance from the keel (baseline) to the center of buoyancy
 $\overline{KG}$ —distance from the keel (baseline) to the center of gravity
 $\overline{KM}$ —distance from the keel (baseline) to the transverse metacenter
 $\overline{KM}_L$ —distance from the keel (baseline) to the longitudinal metacenter
 $\overline{Kg}$ —distance from the keel (baseline) to the center of gravity of a component weight
 k —radius of gyration about an axis, i.e. k_x, k_y
 k —form factor for hull forms, $(C_V - C_F)/C_F$
 k —spring constant
 L —lift force
 L —length, generally
 L —reference length of a ship
 L_{OA} —length of a ship, overall
 L_{OS} —length of a ship, overall, submerged
 L_{PP} —length of a ship between perpendiculars
 L_{WL} —length of a ship on the waterline
 L_w —length of a wave from crest to crest
 LCB —position of longitudinal center of buoyancy
 LCF —position of longitudinal center of flotation
 LCG —position of longitudinal center of gravity
 LWL —load, or design, waterline
 l —length of an internal compartment or tank
 l —longitudinal distance through which a weight is moved
 M —moment, generally
 M —position of transverse metacenter
 M_L —position of longitudinal metacenter
 M_x —bending moment at any section in a ship's length
 MTI'' —moment to change trim one inch
 m —mass, generally, (W/g) or (w/g)
 N —total number of increments in a summation
 NA —neutral axis
 n —revolutions per unit time (generally per second)
 n —any integer
 O —origin of coordinates
 OX —longitudinal axis of coordinates
 OY —transverse axis of coordinates
 OZ —vertical axis of coordinates
 P —power, generally
 P —pitch of a propeller
 P —load force
 $P.C.$ —propulsive coefficient
 PHP —propeller horsepower
 p —pressure, generally
 p_o —static pressure
 Q —torque, generally
 Q_D —torque delivered to the propeller
 Q_o —open-water propeller torque
 Q_R —rudder torque

Q_S —torque measured on the shafting just inside the stern tube
 q —dynamic pressure, $\frac{1}{2}\rho v^2$
 R —resultant force
 R —radius of a propeller
 R —resistance, in general
 R_{AP} —appendage resistance
 R_F —frictional resistance
 R_R —residual (residuary) resistance
 R_T —total resistance
 R_V —viscous resistance
 R_W —wave-making resistance
 R_n —Reynolds number, vL/ν
 r —position vector
 rpm —revolutions per minute
 rps —revolutions per second
 S —wetted-surface area
 SHP —shaft horsepower
 SW —salt water (usually taken as 35 cu ft per ton)
 s_A —apparent slip ratio of a propeller, $1 - v/Pn$
 s_R —real slip ratio of a propeller, $1 - v_A/Pn$
 T —draft
 T_A —draft, aft
 T_F —draft, forward
 T_M —mean draft, $(T_A + T_F)/2$
 T —thrust (force)
 T —period of oscillation, generally
 T —wave period
 T_z —natural period for heaving
 T_θ —natural period for pitching
 T_ϕ —natural period for rolling
 TPI —tons-per-inch immersion
 THP —thrust horsepower
 t —time, generally
 t —transverse distance through which a weight is moved
 t —thickness of a foil section
 t —thrust-deduction fraction, $(T - R_T)/T$
 U —velocity of incident flow
 u —deformation in the x direction
 V —volume, in general
 V —linear velocity in general; speed of a ship
 V_A —speed of advance of a propeller
 V_M —model speed in knots
 V_S —ship speed in knots
 V_x —shearing force across a section at point x along the ship's length
 v —vertical distance through which a weight is moved
 v —specific volume, $1/\rho g$
 v —particle velocity in a gravity wave
 v_M —model velocity in ft/sec
 v_S —ship velocity in ft/sec
 VCB —vertical position of center of buoyancy
 VCG —vertical position of center of gravity

W —weight in general; weight of a ship
 WL —any waterline parallel to the baseline
 WL_1 —changed position of WL
 w —weight of an individual item
 w —Taylor wake fraction, $1 - V_A/V_s$
 w_x —weight per unit length
 w —specific weight, ρg
 x_A —added-mass coefficient
 x —distance from origin along X -axis
 y —distance from origin along Y -axis
 y —distance from neutral axis to a fiber
 y —transverse distance from ship's centerline to center of gravity of flooding water in free communication with the sea
 y —distance from the water surface along an inclined axis
 Z —a point vertically over B , opposite G
 Z —section modulus, I_x/c
 z —distance from origin along Z -axis
 z —heave-motion displacement

Special Naval Architectural Symbols

(Φ) —centerline or centerplane axis
 $(\mathbb{B})$ —baseline
 $\overline{\otimes}$ —midship-section designation
 $\overline{\otimes}_{pp}$ —midlength between perpendiculars
 $\overline{\otimes}B$ —longitudinal distance from amidships to center of buoyancy, B
 $\overline{\otimes}G$ —longitudinal distance from amidships to center of gravity, G
 $\overline{\otimes}F$ —longitudinal distance from amidships to center of flotation, F
 $\overline{\otimes}b$ —longitudinal distance from amidships to the center of the added buoyant layer (mean center of flotation)
 $\overline{\otimes}g$ —longitudinal distance from amidships to the center of gravity of an added or removed weight, w
 ∇ —volume of displacement ("vol"), 35 Δ

Greek Symbols

α (alpha) —angle of incidence; angle of attack
 β (beta) —angle of attack in yaw on the hull (or drift angle), measured from the resultant ship velocity V to the xz_0 -plane
 γ (gamma) —specific gravity
 γ —shearing strain
 Δ (Delta) —displacement in tons salt water
 δ (delta) —small increment
 δ_R —rudder angle
 δT_M —parallel sinkage
 δ_{trim} —change in trim
 ϵ (epsilon) —normal strain
 η (eta) —efficiency, generally

η_B —propeller efficiency behind hull
 η_D —quasi-propulsive coefficient
 η_H —hull efficiency, $(1 - t)/(1 - w)$
 η_O —propeller efficiency in open
 η_P —propulsive efficiency, $\eta_D \times \eta_S$
 η_R —relative rotative efficiency
 η_S —shaft transmission efficiency
 Θ —angle, in general
 θ (theta) —angle of pitch or of trim (about OY -axis)
 Λ (Lambda) —tuning factor, relative to synchronism
 λ (lambda) —linear scale ratio, ship to model
 μ (mu) —Poisson's ratio
 μ (mu) —coefficient of dynamic viscosity
 μ —volume permeability factor
 μ_s —surface permeability factor
 ν (nu) —coefficient of kinematic viscosity; μ/ρ
 ρ (rho) —mass density, weight per unit volume/g
 σ (sigma) —normal stress
 τ (tau) —shear stress
 ϕ (phi) —angle of heel or roll (about OX -axis)
 ψ (psi) —angle of yaw (about OZ -axis)
 ω (omega) —angular velocity
 ω (omega) —circular frequency, $2\pi/T$

Mathematical Symbols

∂ —partial derivative sign
 i — $\sqrt{-1}$
 $\approx$ —is approximately equal to
 $<$ —is less than

$>$ —is greater than
 $\dot{x}$ (one dot over a variable) —the first derivative of the variable
 $\ddot{x}$ (two dots over a variable) —the second derivative of the variable
 $\propto$ —proportional to
 ∞ —infinity
 δ (delta) —a finite increment
 Σ (Sigma) —summation of
 π (pi) —ratio of circumference of circle to diameter
 $\int$ —integral of

Abbreviations for References

ABS —American Bureau of Shipping
 ASNE —American Society of Naval Engineers
 ASCE —American Society of Civil Engineers
 ATTC —American Towing Tank Conference
 BSRA —British Shipbuilding Research Association
 DTNSRDC —David Taylor Naval Ship Research and Development Center
 IMCO —International Maritime Consultative Organization
 ITTC —International Towing Tank Conference
 (R) INA —(Royal) Institution of Naval Architects
 SNAME —Society of Naval Architects and Marine Engineers
 USNA —United States Naval Academy

Introduction to Naval Architecture

CHAPTER 1

Ships Categorized

1-1 Introduction

The forms a ship can take are innumerable. A vessel might appear to be a sleek seagoing hotel carrying passengers along to some exotic destination; a floating fortress bristling with missile launchers; or an elongated box transporting tanks of crude oil and topped with complex pipe connections. None of these descriptions of external appearance, however, does justice to the *ship system* as a whole and integrated unit—self-sufficient, seaworthy, and adequately stable in its function as a secure habitat for crew and cargo. This is the concept that the naval architect keeps in mind when designing the ship and that provides the basis for subsequent discussions, not only in this chapter but throughout the entire book.

In order to discuss naval architecture, it is helpful to place ships in certain categories. For purposes of this text, ships are classified according to their means of physical support and their designed purposes.

1-2 Ships Typed According to Means of Physical Support

The mode of physical support by which vessels can be categorized assumes that the vessel is operating under designed conditions. Ships are designed to operate above, on, or below the surface of the sea, so the air-sea interface will be used as the reference datum. Because the nature of the physical environment is quite different for the three regions just mentioned, the physical characteristics of ships designed to operate in those regions can be diverse.

AEROSTATIC SUPPORT

There are two categories of vessels that are supported above the surface of the sea on a self-induced cushion of air. These relatively lightweight vehicles are capable

of high speeds, since air resistance is considerably less than water resistance, and the absence of contact with small waves combined with flexible seals reduces the effects of wave impact at high speed. Such vessels depend on *lift fans* to create a cushion of low-pressure air in an underbody chamber. This cushion of air must be sufficient to support the weight of the vehicle above the water surface.

The first type of vessel has flexible “skirts” that entirely surround the air cushion and enable the ship to rise completely above the sea surface. This is called an *air cushion vehicle* (ACV), and in a limited sense it is amphibious (figure 1-1).

The other type of air-cushion craft has rigid side walls or thin hulls that extend below the surface of the water to reduce the amount of air flow required to maintain the cushion pressure. This type is called a *captured-air-bubble vehicle* (CAB). It requires less lift-fan power than an ACV, is more directionally stable, and can be propelled by water jets or supercavitating propellers (figure 1-2). It is not amphibious, however, and has not yet achieved the popularity of the ACVs, which include passenger ferries, cross-channel automobile ferries, polar-exploration craft, landing craft, and riverine warfare vessels.

HYDRODYNAMIC SUPPORT

There are also two types of vessels that depend on *dynamic support* generated by relatively rapid forward motion of specially designed hydrodynamic shapes either on or beneath the surface of the water. A principle of physics states that *any moving object that can produce an unsymmetrical flow pattern generates a lift force perpendicular to the direction of motion*. Just as an airplane wing (airfoil) produces lift when moving through the air, a *hydrofoil*, located beneath the surface and attached by means of a surface piercing strut, can dy-



Figure 1-1. A one-hundred-ton surface effect ship, an air-cushion vehicle (ACV), making a test run. These vessels are supported on a cushion of relatively low-pressure air supplied by ducted internal fans. They are propelled at potential speeds up to 80 knots by turbine-driven water jets. (Courtesy of Bell Aero Space Co.)

namically support a vessel's hull above the water (figure 1-3).

Planing hulls are hull forms characterized by relatively flat bottoms and shallow V-sections (especially forward of amidships) that produce partial to nearly full dynamic support for light displacement vessels and small craft at higher speeds (figure 1-4). Planing craft are generally restricted in size and displacement because of the required power-to-weight ratio and the structural

stresses associated with traveling at high speed in waves. Most planing craft are also restricted to operations in reasonably calm water, although some "deep V" hull forms are capable of operation in rough water.

HYDROSTATIC SUPPORT

Finally, there is the oldest and most reliable type of support, *hydrostatic support*. All ships, boats, and primitive watercraft up to the twentieth century have de-



Figure 1-2. Another type of surface effect ship, a captured-air-bubble vehicle (CAB), is supported on a cushion of air that is screened at the sides by the rigid side structure of the vessel.



Figure Figure 1-3. The hydrofoil-supported vessel. These craft have the best seakeeping potential of the various hydrodynamically supported vessels.

pendent upon the easily attained buoyant force of water for their operation.

This hydrostatic support, commonly recognized as *flotation*, can be explained by a fundamental physical law that the ancient philosopher-mathematician Archimedes defined in the second century B.C. *Archimedes' Principle* states that a body immersed in a liquid is buoyed up (or acted upon) by a force equal to the weight of the liquid displaced. This principle applies to all vessels that float (or submerge) in water—salt or fresh. And from this statement the name of the ships in the category are derived; they are generally called *displacement hulls*.

Although this ship type is very familiar, its subcategories warrant special discussion. For example, in some vessels reasonably high speed must be combined with the ability to carry light cargo or to move more comfortably in rough water than a planing hull. High-speed planing-hull characteristics can be modified to produce a semidisplacement hull or semiplaning hull (figure 1-5). These compromise craft, of course not as fast as full-planing hulls but faster than conventional displacement hulls, must have more power and less weight than the latter. Such types are obviously the result of “trade-offs.”

The example cited above lies between clear-cut physically defined categories—it is not a good example of a variation of a true displacement-type ship. The latter must be recognized primarily as a displacement vessel, and its variations depend primarily on the distribution of buoyant volume—the extent of the depth and breadth of the hull below the water.

The most ubiquitous type of displacement ship can be generally classified as the *common carrier*, a seagoing vessel. It may be employed for passenger service, light cargo-carrying, fishing by trawling, or for hundreds of other tasks that do not require exceptional capacity, speed, submergence, or other special performance (figure 1-6). It is the most common and easily recognizable type of ship, with moderate displacement, moderate speeds, moderate to large lengths, and moderate capacities. It usually embodies the maximum in cruising range and seaworthiness. It is the “ship for all seasons.” It is the standard to which all other ship classifications in the displacement category may be referred (figure 1-7).



Figure 1-4. HMS *Tenacity*, a fast fisheries patrol vessel, is essentially a planing hull. A large example of this type, she is 142 feet long, powered by gas turbines, and capable of 40-kt speeds. She was originally intended as a fast surface-to-surface missile carrier. (Courtesy of Vosper-Thornycroft, Ltd.)



Figure 1-5. The USS *Welsh*, a fast navy gunboat, is capable of planing speeds but has a combination power plant more economical for cruising at lower speeds. She thus operates as a semiplaning hull.



Figure 1-6. The handsome and conventional displacement ship is an excellent example of a modern luxury ship designed as a passenger cruise ship, one of the last examples of a disappearing type, the seagoing passenger ship. (Courtesy of Swan-Hunter Shipbuilders, Ltd.)



Figure 1-7. The hard-working and valuable displacement ship of the SL 7 type is currently the fastest cargo ship in the world, capable of sustained cruising speeds of 33 knots. (Courtesy of Sea-Land Industries, Inc.)

The closest relative to this standard vessel, which plays a crucial role not only in world commerce but in the survival of the industrial world as well, is the bulk oil carrier, the tanker, or supertanker (figure 1-8). These terminologies are common but unspecific, and in this discussion they are inadequate, for what was called a supertanker several years ago is today not a supertanker. The industry itself has created a far more explicit nomenclature. Based upon the index of 100,000 tons oil cargo capacity, the size categories are LCC (Large Crude Carrier), VLCC (Very Large Crude Carrier), and ULCC (Ultra Large Crude Carrier). Any tanker greater than 100,000 tons but less than 200,000 is a LCC, those between 200,000 and 400,000 are VLCCs, and those over 400,000 are ULCCs. The current necessity for these designations becomes clear when we realize that before 1956 there were no tankers larger than 50,000 tons, and not until the early sixties were any ships built larger than 100,000 tons. In 1968 the first ship over 300,000 tons was built. With their bulk and enormous capacity (four football fields can be placed end to end on one of their decks), these ships are designed and built to be profit-makers, enormously long,

wide, and deep, carrying thousands of tons of crude oil per voyage at the least cost. Few of these elephantine tankers have more than one propeller shaft or rudder. Their navigation bridges are nearly one quarter of a mile from their bows. Their top service speed is so low that a voyage from an Arabian oil port to a European destination normally takes two months.

Such vessels belong to a category of displacement ship that has a great range of buoyant support. They have a very large and disproportionate hull volume below the surface when fully loaded. Indeed, the cargo weight far exceeds the weight of the ship itself. The draft or depth of water required for a fully loaded VLCC runs to 50 or 60 feet and the ULCC may be 80 feet. Such ships belong in the exclusive category of displacement vessels called *deep displacement* ships.

There exists another type of displacement hull with extreme draft. However, its similarity to the crude-oil carrier of the preceding discussion goes no further than that. This type of vessel is called the SWATH (Small Waterplane Area Twin Hull). Briefly, this rather rare breed of ship is designed for relatively high speed and stable platform in moderately rough water. Its future

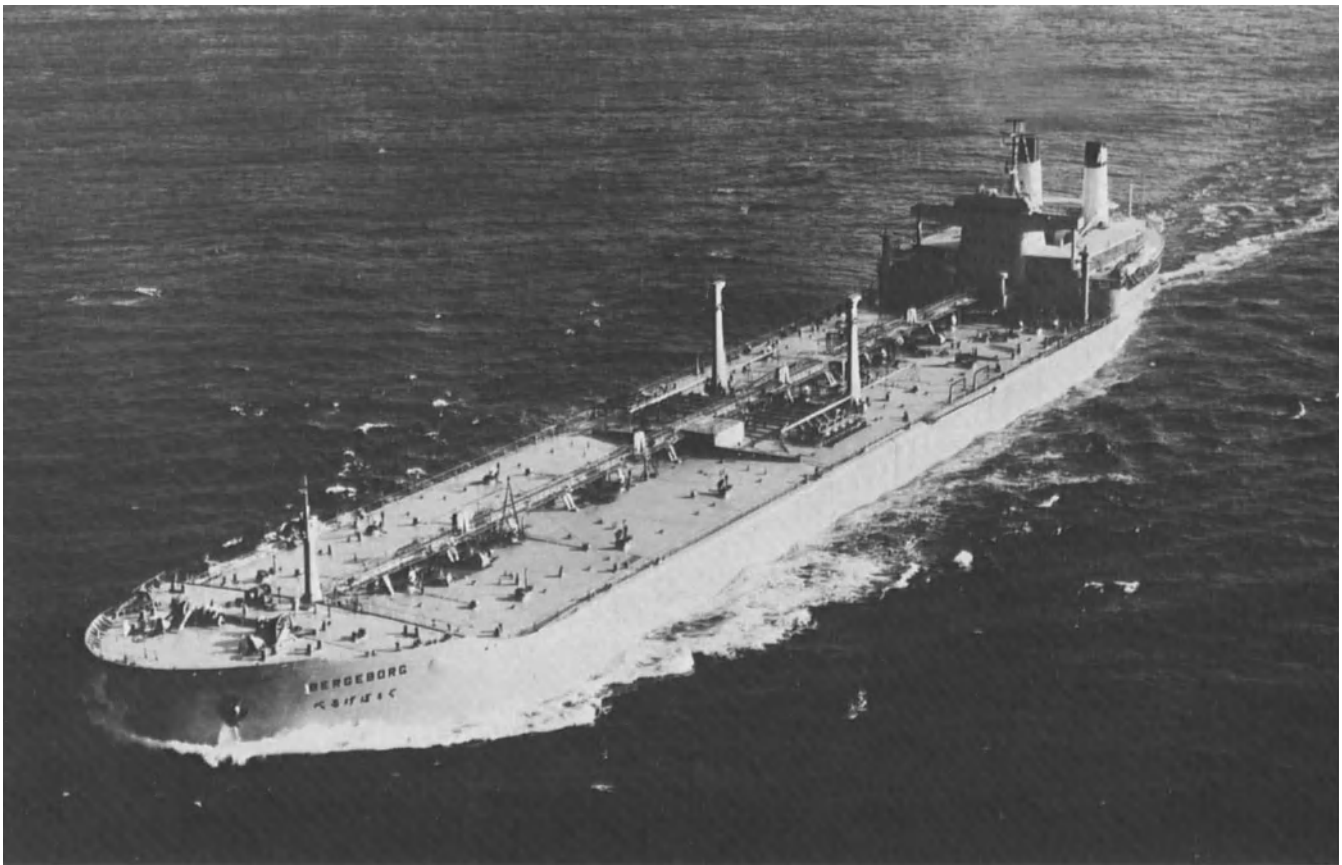


Figure 1-8. The heaviest and largest of all displacement ships, the crude oil tanker. Sometimes referred to as Large Crude Carriers and Very Large Crude Carriers, such ships are built as large as one-half-million tons capacity.

is problematical, but the theory of placing the bulk of the displacement well below the surface and extending the support to the above-water platform or deck through the narrow waterline fins or struts is sound. Twin hulls connected by an upper platform provide the necessary operating stability (figure 1-9).

The most significant class of displacement hull for special application is the submarine, a vessel for completely submerged operation. The nature of the submarine and a description of her various operational attitudes, both static and dynamic, is covered in subsequent chapters. It is only necessary here to emphasize that submersible vessels are specifically displacement vessels applying the theory of Archimedes' Principle and all that it implies (figure 1-10).

MULTIHULL VESSELS

There is one other type of hull in common use that has not yet been mentioned, primarily because it fits into none of the categories described but rather can exist comfortably in any. This craft is the so-called *multihull*

vessel—the catamaran and the trimaran. These vessels are most frequently displacement hulls in their larger sizes, such as the SWATH mentioned above, or more conventionally, ocean research vessels requiring stable platforms and protected areas for launching equipment (figure 1-11). There are also the twin-hulled CAB vessels mentioned earlier and high-speed planing catamarans. Actually, the multihull ship is an adaptation of any of the basic hull categories to a special application that requires exceptional transverse stability and/or the interhull working area.

Figure 1-12 indicates the body profiles (with no relative scale) that have just been described and relates them to their means of physical support. They are arranged from high to low speed, except for the multihull types, which may be either speed, depending upon their purpose.

1-3 Other Criteria

There are other criteria that justify the widely varied configurations of ship design. They are the result of

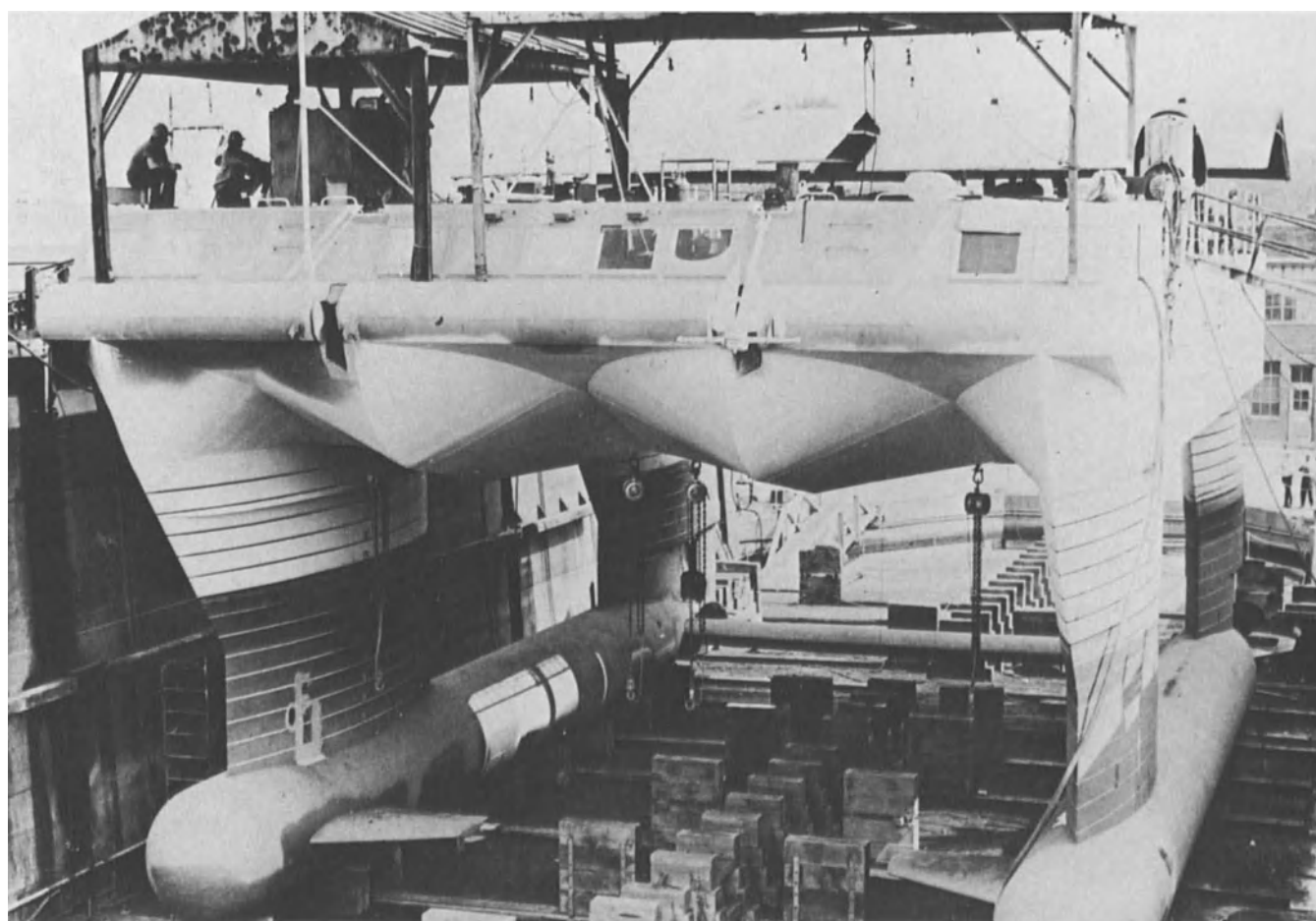


Figure 1-9. The deep-draft types are still largely experimental concepts. They are categorized as small waterplane area twin-hull designation, or SWATH. Potentially fast for displacement ships, they are very stable with minimum motion in a seaway.

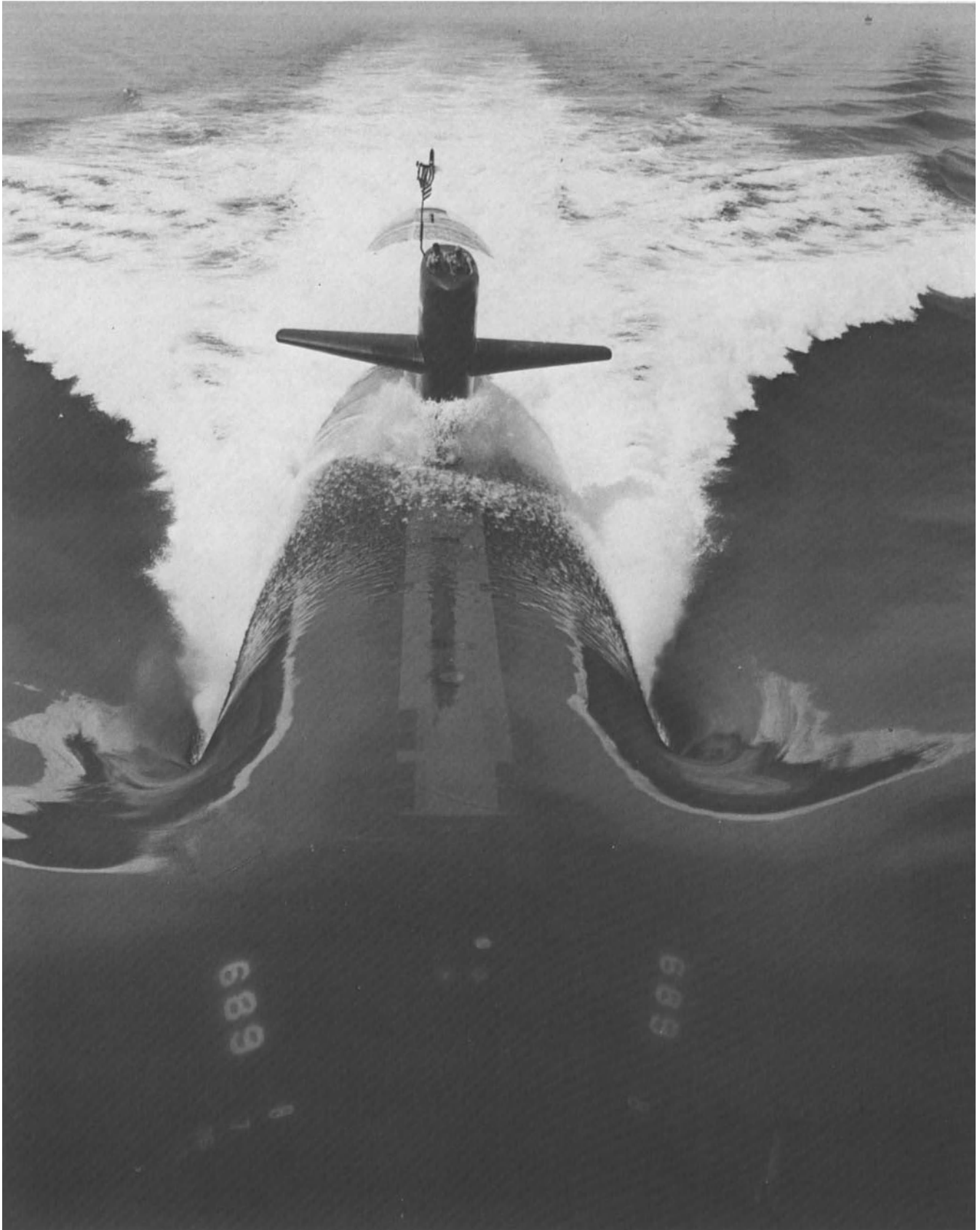


Figure 1-10. The modern navy attack submarine, the best example of an advanced submersible vessel. Nuclear powered, they are designed for optimum performance entirely below the surface.



Figure 1-11. The oceanographic research vessel USNS *Hayes*, one of the best examples of a multi-hull vessel, generally a costly, controversial type

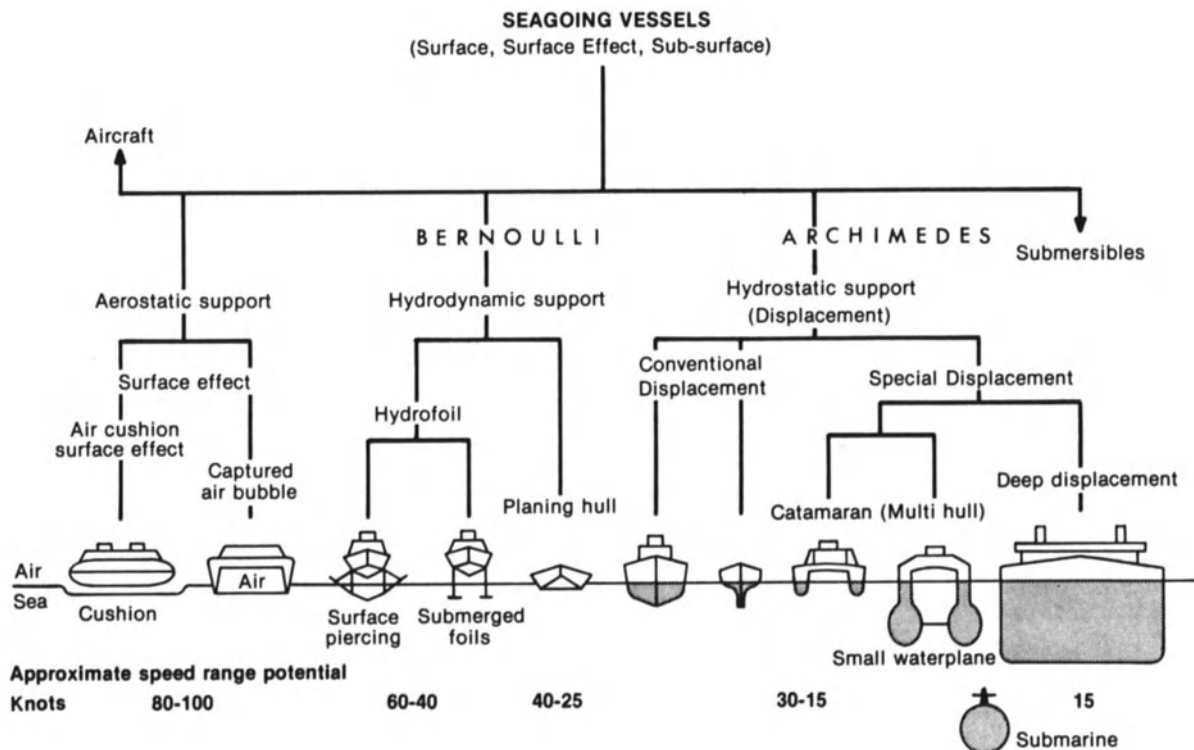


Figure 1-12. Categories of seagoing vessels arranged according to their mode of support on or in the sea

trade-offs concerning cost, mission, speed, endurance, payload (cargo or weapons capacity), operating environment (stability, survivability, and port requirements), reliability, appearance, personal comfort and habitability, and political considerations. The relative importance of the various factors is dictated by the purpose of the vessel, which is set by the commercial firm, government, or individual who purchases the vessel. A useful classification based on purpose includes the following categories: merchant and commercial ships, naval vessels, and pleasure craft.

MERCHANT AND COMMERCIAL SHIPS

Merchant and commercial ships are generally bought to earn a profit. The previously discussed cargo ships are designed for the minimum (or at least competitive) “required freight rate,” which involves predicting the “life-cycle cost” of the ship, including the acquisition costs, the operating and maintenance costs, and any salvage value remaining when the ship is sold. A “cash-flow analysis” is made to establish what rate of return can be expected on the owner’s investment.

New designs of all commercial vessels, including cargo ships, passenger liners, fishing boats, offshore supply vessels, and tugboats, must compete economically with similar vessels available from the many worldwide shipbuilders. Government subsidies protecting the national shipbuilding industry from foreign competition can result in lower costs to the purchaser, even when the actual ship construction costs are higher. Thus, political

considerations can play an important role in the economics of commercial ship design and construction.

Appearance, personal comfort, and reliability are necessary for a luxury passenger liner to attract customers, whereas payload, endurance, and ability to survive a hostile sea environment are important considerations in the design of fishing vessels. Offshore supply vessels are concerned with speed for oil-rig crew transport or emergency services, but slower speeds may be acceptable when payloads such as drill pipe and drilling mud are the principal cargo. Operating environment includes both wind and wave conditions at sea and port and harbor capabilities ashore. Thus, deep-draft vessels may be excluded from certain geographic areas. Special-purpose cargo-handling devices such as the unloading ramps on roll-on/roll-off (Ro/Ro) ships (figure 1–13) may be necessary for quick turnaround both at principal worldwide ports and those of underdeveloped countries. The latter ports impose other cargo-handling restrictions on the ship designer.

NAVAL AND COAST GUARD VESSELS

Naval vessels are generally classified as combatants or auxiliaries, although there are special-purpose craft that do not fit easily into either category. For large combatant warships such as aircraft carriers, guided-missile cruisers, destroyers, and nuclear submarines, all of the previously mentioned factors become important—hence the enormous cost of such ships. Their military mission is of prime importance, but carrying out the mission



Figure 1–13. The *Lipscomb Lykes*, a Ro/Ro ship. (Courtesy of Melvin Freden.)

depends on speed, endurance (possibly aided by at-sea replenishment for surface ships), weapons payload, and ability to operate and survive in hostile environments. Reliability under combat conditions, the appearance of military power, crew habitability that influences reenlistments, and the political importance of who becomes the prime contractor and principal weapons-system subcontractors: all these are factors that must be taken into consideration, making the construction and operation of warships very expensive for taxpayers.

Naval auxiliaries are more closely related to commercial ships in appearance, but their mission may involve operating with warships, which requires compatibility in terms of speed, endurance, required payload, and the ability to conduct replenishment operations during poor sea conditions. Thus, one can expect the cost of such ships to be greater than that of their commercial counterparts.

Oceanographic research vessels, Coast Guard cutters, and ice-breakers all have missions in which endurance, reliability, ability to operate in difficult environments, and habitability are important. Since smaller vessels have limited fuel capacity, there is a trade-off between speed and endurance; hence two types of power plant are frequently used to optimize both speed and endurance. The more exotic craft discussed in the previous sections generally sacrifice payload and endurance for speed.

PLEASURE CRAFT

Pleasure craft, both motor powered and sail powered, come in a wide variety of sizes and shapes to suit individual requirements and tastes. The economic trade-offs are based on what the potential buyer can afford or thinks he can afford. Appearance, speed, personal comfort and habitability, and stability are the major criteria for designs that satisfy the purpose of the craft, which is the enjoyment of leisure time.

GENERAL CONSIDERATIONS

In such general comparisons and categorizations it is necessary to return to a practical perspective. It is all very well to arrange type categories with respect to the common denominator of supporting force or mission, but the question of relative significance in harder terms must ultimately be answered. The graphic comparisons in figure 1-14 convey the great variation in relative performance and capability as a function of ship category based on type of support.

How many ships of each of these categories can justify themselves in terms of economic support and environmental capabilities? How many are purely experimental? What can be expected of their future? Where these questions apply to the problems faced by the ship designer, an attempt has been made in the following chapters to provide the background for adequate evaluation.

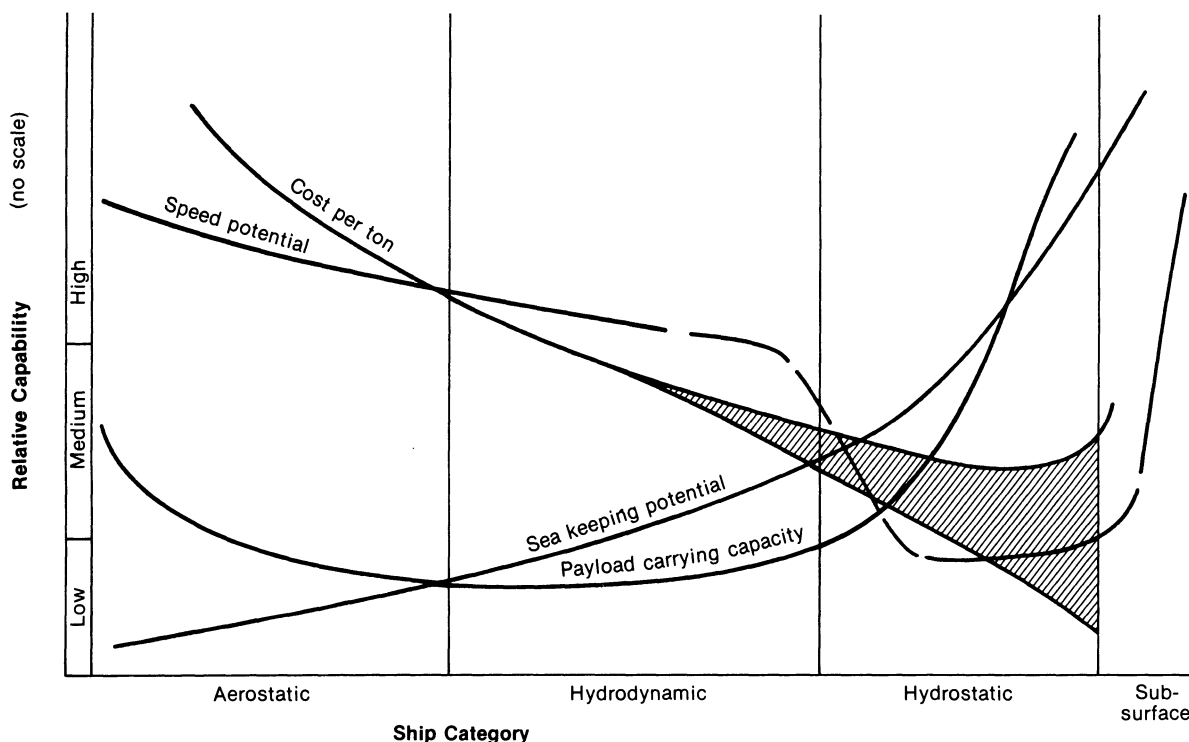


Figure 1-14. The relative performance and associated capabilities of the ship categories of figure 1-12

A more detailed discussion of these comparative factors can only be made after the technical presentations of this book have been absorbed. But it must be emphasized here that most of this book will deal with the physical nature of *displacement* ships, simply because almost all of the ships on the world's oceans are and probably will be of this type. They carry the raw materials of world commerce and a nation's military strength to most parts of the globe. Without them the civilized industrialized world would quickly collapse.

The ships of recent years have noticeably progressive features in their external configuration. The old stacks or funnels have been replaced by raked, streamlined stacks or transverse pairs of slim diesel funnels. Superstructures have become crisp and uncluttered. Hulls of tankers and bulk carriers have become monstrous in volume. Fast cargo carriers and naval vessels have acquired new grace in their sheer and flare. Below the waterline, improved hydrodynamic knowledge has resulted in bulbous forefoot extensions and improved rudder configurations. There are a multitude of internal developments provided by modern technology, including the less visible changes in strength and performance allowed by improved metals and other materials.

1-4 The Systems Approach

The greatest change in new ships, however, is not very evident in their structure. This is because designers, planners, and operators recognize that a ship is an extremely complex but integrated total *system* (figure 1-15).

It is increasingly difficult to design and build a ship without regard to the systems-engineering approach. Because of the rapidly mushrooming technology of this century, there has been growing specialization within

the engineering professions. This has led to the need for a way to deal with complex assemblies made up of many specialized components. If they are to be capable of optimum performance, such complex assemblies as the Trident submarine or the nuclear aircraft carrier must be designed in an orderly manner. This integrated approach is ordinarily referred to as *systems engineering*.

Systems engineering is employed in the design of all naval vessels and most commercial craft today, and the student of ship design should become familiar with it early in his engineering education. We might define the approach as a process for *achieving significant objectives, allocating resources, and organizing information* so that all major aspects of a problem can be precisely determined and coordinated according to a plan. Systems engineering supplies the bridge between what is needed and what is technically feasible.

SYSTEMS IN SHIPS

Systems engineering, whether it is applied to a large ocean transport ship, a warship, or a very small vessel, implies total integration of all subsystems to provide a functional unit that achieves the basic mission of the ship. This means that ship control must function through the internal and external communications systems, and the machinery and propulsion systems must react to control, signaling their responses on display instruments at the central control station. The weapons systems of a warship must function on order with simultaneous execution and respond to all safety and protective systems. Systems engineering includes all automatic control systems as well as a multitude of engineering and electronic subsystems that maintain order and perform daily living and emergency functions. In the last century



Figure 1-15. The USS *Enterprise* (CVAN 65), the largest of all ship systems

or more of successful mechanical propulsion, the ship has undergone fundamental changes; no longer is she merely a large floating vessel with a relatively isolated power plant, isolated cargo holds and living quarters, and a lonely navigation bridge with its crude mechanical or sound-signaling device to the engine room. In a sense, the ship of a century ago was a system too, but her design lacked the systematic, integrated approach demanded for the successful modern ship.

In the design of a modern warship, shipbuilders and naval architects recognize the following major subsystems of functional groupings:

1. Hull
2. Machinery and propulsion plant
3. Electric plant
4. Command and surveillance
5. Auxiliary systems
6. Outfit and furnishings
7. Armament

The interfaces and interplay among these subsystems must be identified and defined. The secondary subsystems in each must then be defined by more detailed functional blocks. Components for emergency operation and casualty procedures must be provided, and in the design stages particularly, prime consideration must be given to weight and buoyancy, speed, power and endurance, volume relationships, payload capacity, general arrangements and habitability, ship-motion effects, and acoustic influences. Integration of these multiple considerations is generally accomplished before the actual preliminary design is begun.

It has been suggested that an integrated ship has a tendency to become an inflexible black box, with no capacity for growth. If this were true, the ship would contain components that might be incompatible with the logistics plans supporting other ships, and the defects in such resulting ships could be located and eliminated in the design process. Actually, the integrated ship can be made *more* rather than less compatible with logistics by using the maximum of standardized components and adapters for special applications in the subsystem design. The command control subsystem, for example, requires multipurpose, interchangeable displays tied together at the central computer subsystem and properly programmed to achieve the required compatibility and flexibility. It must also be recognized that in the "whole" ship design, proper systems engineering provides for standardization, interchangeability, and central procurement of unitized components.

THE EFFECTIVE SHIP'S SYSTEM

From the preceding section, one might get the impression that nowadays all modern ships are designed and

built as completely integrated systems. This is not completely true; rather, systems integration is a goal to be achieved insofar as economics allow, demands require, and skills of the designers and builders enable. In other words, all ships are systems, but the integration of their subsystems varies. Their usefulness in fulfilling their missions depends upon the degree of integration. The measure of this degree is found in the science of *systems effectiveness*. While it is not the purpose of this text to discuss this science in detail (most engineering students encounter the study of systems effectiveness early in their engineering curriculum), it is helpful to mention that the designer should expose his ship's system design to this orderly evaluation procedure.

Because of the great number of ship types, the varying complexities of their designs, and their nearly infinite range requirements, it is not possible to standardize an approach to effective system design. Simply stated, systems effectiveness is defined as the probability that the system will operate successfully under specific conditions throughout a given time period. This definition is general and can be applied to any system, including the ship itself.

While actual operational systems effectiveness extends beyond the designer's control and contains at least two other factors—human decisions and responses—external to design, the designer is the initial creator of the system and its effectiveness.

Some examples of engineering integration may illustrate what is *not* an effective system. A whole system comprised of determinate subsystems cannot be effective where there are such things as overlapping requirements, incompatibility, mechanical deficiencies, and lack of environmental adaptability. For example, the propulsion unit of a high-speed planing hull, if it is a marine propeller, must have comparatively small diameter, high pitch, and rapid revolutions per minute to be compatible with the hull system. To be compatible with its environment, the propeller must be made of a metal that will not form a terminal of an electrolytic system and that will therefore protect against electrolysis. The designer must ask himself such questions as: Does subsystem *A* have a material reliability of 50 percent when subsystem *B*—which is dependent on subsystem *A*—has a reliability of 95 percent and the total system requirement is to be 90 percent reliable? Further, are associated subsystems compatible? Are there gasoline auxiliary generators in a system where the primary fuel is diesel oil?

From these simple examples of systems-effectiveness evaluation, it is but a short step to realizing the need for full evaluation of the many systems existing in a modern, complex ship. Such responsibilities for the U.S. Navy lie with the Naval Sea Systems Command.

CHAPTER 2

Engineering Fundamentals in Ship Design

2-1 Introduction to Mechanics

Newtonian mechanics is the study of the motion of objects and bodies, both solid and fluid. That portion of mechanics applicable to ship design views solid bodies and fluids as continuous media. Although many of the analytical laws of motion were originally developed for particles and rigid bodies, they can be considered relevant to continuous media when applied to the center of mass or center of gravity. Isaac Newton (1642–1727) formulated the basic laws of motion when he postulated that there exists a frame of reference for which

1. Every body continues in its state of rest or of uniform motion in a right [straight] line, unless it is compelled to change that state by forces impressed upon it.
2. The change of motion [i.e., mechanical momentum] is proportional to the motive force impressed; and is made in a direction of the right [straight] line in which that force is impressed.
3. For every action there is always opposed an equal reaction; or, the mutual actions of two bodies upon each other are always equal, and directed to contrary parts.

The postulated frame of reference is called an *inertial frame*, and the commonly accepted inertial frame of reference is the fixed stars. For most engineering work at moderate speeds near the earth's surface, the earth itself can be considered a satisfactory frame of reference.

Newton's second law is commonly formulated as

$$\Sigma \mathbf{F} = \frac{d}{dt} (m\mathbf{v}) \quad (2-1)$$

where $\mathbf{F}$ = force, m = mass, $\mathbf{v}$ = velocity.

(Note the use of bold-faced type to indicate a vector quantity.)

In words, the second law can be formulated as follows: *The summation of forces on an object equals the time rate of change of momentum of the object.* For rigid bodies, this reduces to

$$\Sigma \mathbf{F} = m\mathbf{a} \quad (2-2)$$

where $\mathbf{a}$ = acceleration.

These are *vector* equations, since force systems and momentum changes involve both magnitude and direction. More about the nature of vector quantities will be stated in the next section.

Mechanics can be further subdivided into the following categories:

1. Statics—The study of situations for which the summation of both forces and moments is equal to zero. This special case includes bodies at rest and in uniform, unaccelerated motion.
2. Kinematics—The study of the geometry of motion.
3. Dynamics—The study of the forces and moments causing and resulting from motion.

Engineering students generally take separate courses covering various aspects of mechanics. These courses may include

1. Statics—The study of forces and moments on rigid, unaccelerated objects. A brief introduction is included in this chapter.
2. Dynamics—The study of the effects of forces and moments on the kinematics of rigid body motion. Applications involving ship motions and maneuverability are covered in chapters 13 and 14.
3. Material science—The study of the properties of materials. This is briefly summarized for ship applications in chapter 4.
4. Solid mechanics or strength of materials—The study of the external and internal forces associated with the

deformation of solid structures. This is summarized for ship applications in chapter 5.

5. Fluid mechanics (both hydrostatics and fluid dynamics)—The study of the forces and motions associated with substances that cannot sustain shear forces when in static equilibrium. (Fluids deform continuously when subjected to shear stresses.) The principles of the conservation of mass and energy are added to the basic laws of motion for such studies. The subject of hydrostatics is summarized in this chapter. Applications of hydrostatics to ship design are covered in chapter 3 and in chapters 6 to 9. Hydrodynamics is covered in chapters 11 to 14.

2-2 Introduction to Statics: Vectors and Scalars

As stated in the previous section, statics is the study of situations for which the summation of both forces and moments is equal to zero. Since both forces and moments represent vector quantities, those aspects of vector algebra that apply to summations will be reviewed. But first it is essential that two quantities be defined.

A *scalar* is a quantity, completely specified by a number on an appropriate scale, such as mass, length, time, or temperature. It is independent of the choice of a coordinate system.

A *vector* is a quantity completely specified by a magnitude and a direction, such as force, velocity, or acceleration. It is generally represented by a directed line segment or arrow located somewhere in a coordinate

system and is identified in this text by bold-faced letters. If the coordinate system is based on an inertial frame of reference, the magnitude and direction are fixed quantities, but if the coordinate system is moving relative to an inertial frame of reference, the magnitude and direction of the vector may depend upon the motion of the coordinate system, which is itself a vector quantity. A typical example of this phenomenon is the observation of relative motion from a moving vehicle. Velocities perpendicular to the direction of motion appear the same to observers in both frames of reference. However, velocities parallel to the direction of motion appear as *relative* velocities to the moving observer and must be corrected for the motion of the moving frame of reference if *absolute* magnitudes and directions are desired.

Although this distinction will become important in later chapters that treat the dynamic nature of ship motions, the study of statics does not involve moving frames of reference.

MAGNITUDE AND DIRECTION

The magnitude of a vector is always a positive quantity and corresponds to the length of vector represented graphically. For example, local wind speed corresponds to the length of a wind vector on a diagram of a storm system, or the particle velocities in a breaking wave are represented by an array of vectors as illustrated in figure 2-1, in which the length of the vector represents the

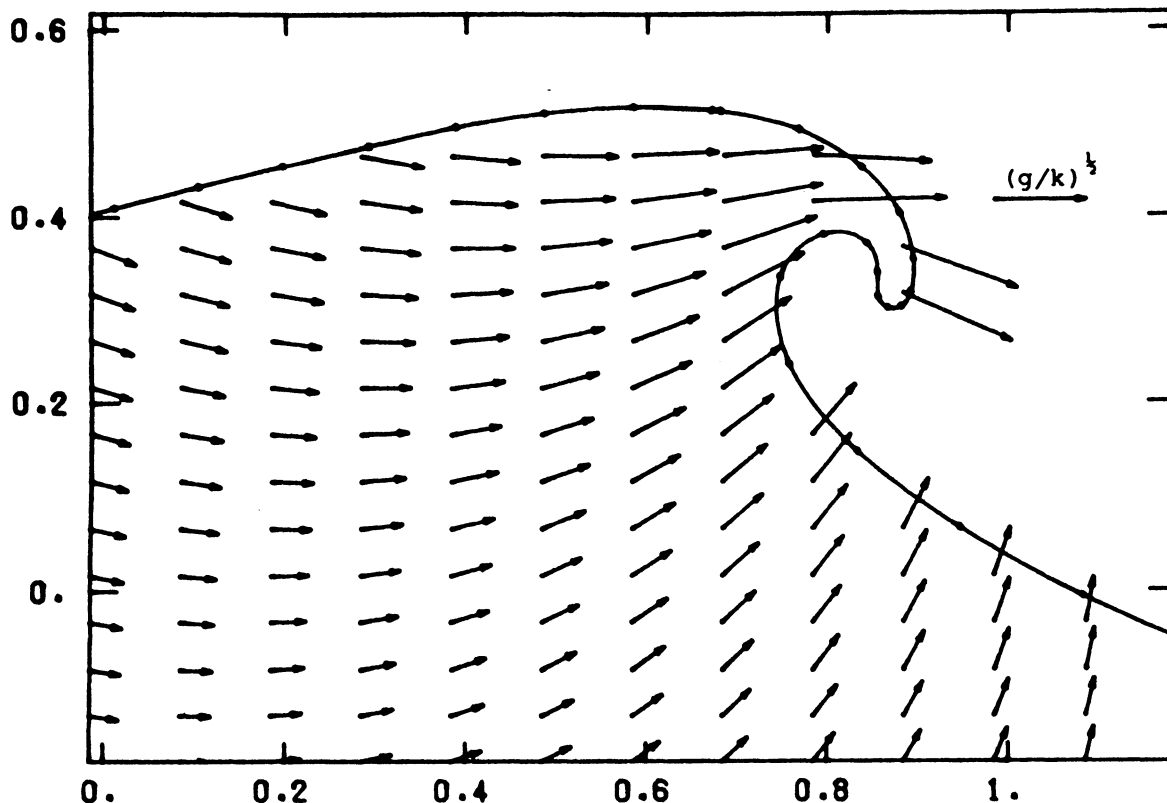


Figure 2-1. Particle velocities for a breaking wave showing the plunging jet. (From Cokelet 1978.)

speed of the particles at a particular location in the breaking wave (Cokelet 1978).

The magnitude of a vector is usually indicated by using an absolute value sign, but it may also be indicated by eliminating the boldface.

$$\text{magnitude of vector } \mathbf{A} = |\mathbf{A}| = A \quad (2-3)$$

When a vector quantity $\mathbf{a}$ is multiplied by a scalar quantity m , as in equation 2-2, the direction of both $\mathbf{a}$ and the resultant scalar product $\mathbf{F} = m\mathbf{a}$ are the same. The magnitude of $|\mathbf{F}|$ equals the magnitude of $|\mathbf{a}|$ multiplied by m . The vector $-\mathbf{F}$ is defined as a vector with magnitude equal to $\mathbf{F}$ but having an opposite sense to $\mathbf{F}$. The direction of a vector is indicated by the angle its representing arrow makes with a coordinate-system axis, or graphically by the arrow itself. It may be defined analytically by the coordinates of the head of the arrow when the tail is placed at the origin of the set of coordinates.

ADDITION AND SUBTRACTION OF VECTORS

The summations of forces and moments involve the addition and subtraction of vectors and may be accomplished either graphically or analytically. The graphical method involves the use of the parallelogram law based on the commutative property of vectors.

To illustrate vector addition and subtraction, let $\mathbf{A}$ and $\mathbf{B}$ be two vectors represented by the two arrows $\mathbf{A}$ and $\mathbf{B}$ in figure 2-2, which in turn represent the directed

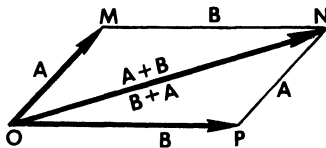


Figure 2-2. Vector addition

line segments $\mathbf{OM}$ and $\mathbf{OP}$. The sum $\mathbf{A} + \mathbf{B}$ is represented by the diagonal of the parallelogram $\mathbf{OMNP}$, which also demonstrates the commutative law for vector addition

$$\mathbf{A} + \mathbf{B} = \mathbf{B} + \mathbf{A} \quad (2-4)$$

Figure 2-3 illustrates the head-to-tail addition method

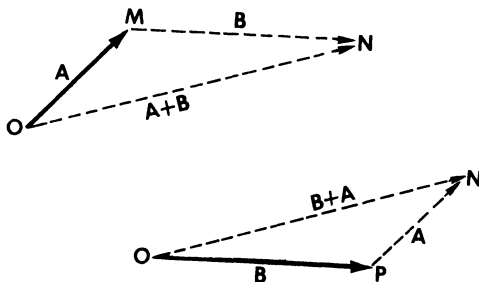


Figure 2-3. Commutative law (triangle rule)

(triangle rule) that may be employed to add several vectors using the commutative law.

The associative law of vector addition,

$$(\mathbf{A} + \mathbf{B}) + \mathbf{C} = \mathbf{A} + (\mathbf{B} + \mathbf{C}) \quad (2-5)$$

is demonstrated in figure 2-4 using the head-to-tail addition method. If the $\mathbf{A} + \mathbf{C}$ parallelogram were added to the diagram, a total of six different sequences for the addition of the three vectors could be followed, all giving the same result.

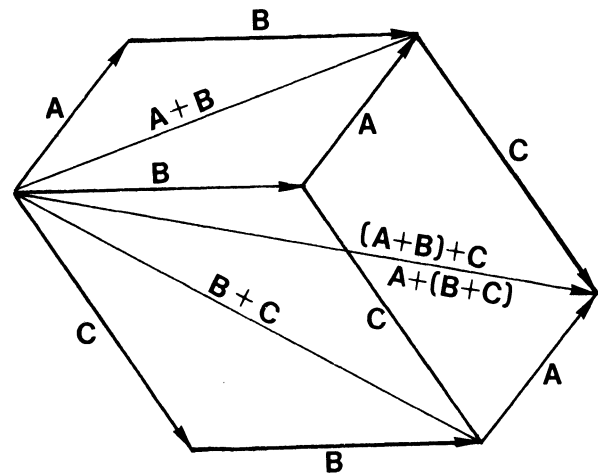


Figure 2-4. Associative law

Vector subtraction is accomplished by adding a negative vector using head-to-tail addition or by constructing the parallelogram using the appropriate negative vector as illustrated in figure 2-5. As can be seen from

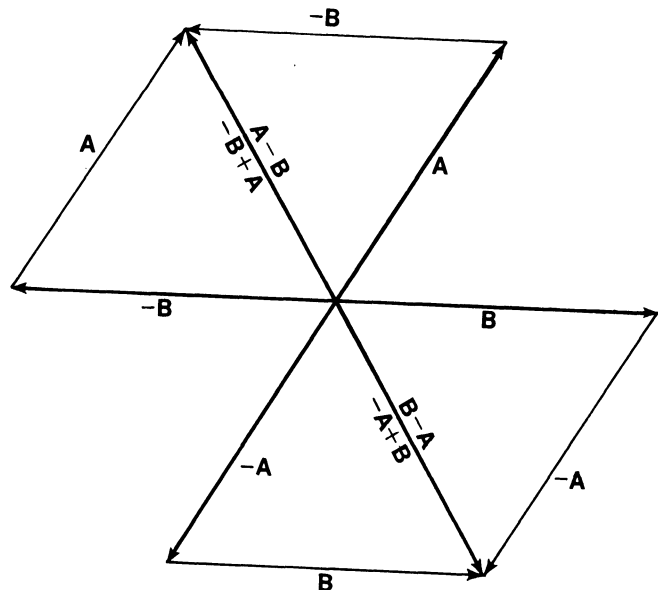


Figure 2-5. Vector subtraction

the diagram,

$$\mathbf{A} - \mathbf{B} = -(\mathbf{B} - \mathbf{A}) \quad (2-6)$$

equations 2-4, 2-5, and 2-6 have a meaning that is slightly different from the customary one because they are vector equations and require the magnitude as well as the direction of the left-hand side to be equal to that of the right-hand side. Two vectors can be said to be equal only if their directions as well as their magnitudes are equal to each other.

The analytical method for vector summations involves the resolution of the vectors into components along coordinate axes. This method has a distinct advantage over the graphical method for problems in three (or more) dimensions. Arbitrary vectors could be used for the development of the analytical method for vector summations, but since much of the text involves the analysis of forces, their resolution into components will be used as the example.

2-3 Resolution of Forces into Components

The most common method for the resolution of forces into components involves the use of an orthogonal coordinate system with mutually perpendicular axes, such as the Cartesian coordinate system illustrated in figure 2-6. In this case

$$\mathbf{F} = \mathbf{F}_x + \mathbf{F}_y + \mathbf{F}_z$$

and

$$|\mathbf{F}| = \sqrt{|\mathbf{F}_x|^2 + |\mathbf{F}_y|^2 + |\mathbf{F}_z|^2} \quad (2-7)$$

where $\mathbf{F}_x$, $\mathbf{F}_y$, and $\mathbf{F}_z$ are the rectangular components of $\mathbf{F}$ obtained by projecting $\mathbf{F}$ onto the x -, y -, and z -axes.

It is useful at this point to introduce the concept of the unit vectors $\mathbf{i}$, $\mathbf{j}$, and $\mathbf{k}$ of magnitude 1, directed

respectively along the positive x -, y -, and z -axes. This will enable the rectangular components F_x , F_y , F_z to be treated as scalars multiplied by the unit vectors $\mathbf{i}$, $\mathbf{j}$, and $\mathbf{k}$.

$$\mathbf{F}_x = F_x \mathbf{i} \quad \mathbf{F}_y = F_y \mathbf{j} \quad \text{and} \quad \mathbf{F}_z = F_z \mathbf{k}$$

Thus,

$$\mathbf{F} = F_x \mathbf{i} + F_y \mathbf{j} + F_z \mathbf{k} \quad (2-8)$$

as shown in figure 2-7. The vector components of $\mathbf{F}$, $\mathbf{F}_x$, $\mathbf{F}_y$, and $\mathbf{F}_z$ can now be treated as scalar components F_x , F_y , and F_z , which may be positive or negative, depending on the sense of the vector components.

The actual computation of the magnitude of the force components is much simpler in two dimensions than in three dimensions, because only one angle is involved. Since most of the problems to follow will involve forces in a two-dimensional plane, the remaining discussion of vector forces will be limited to the resolution of $\mathbf{F}$ into $\mathbf{F}_x$ and $\mathbf{F}_y$ components.

The placement of coordinate axes is always arbitrary, and one can change their orientation to a more convenient one at the beginning of any problem. Consider a force $\mathbf{F}$ that lies in the plane of the paper as illustrated in figure 2-8. By changing the x - and y -axes from their arbitrary orientation to another orientation, such as the x' - and y' -axes, $\mathbf{F}$ can be made equal to the addition of many different sets of component vectors $\mathbf{F}_x'$ and $\mathbf{F}_y'$.

If Θ represents the angle between the positive x -axis and $\mathbf{F}$ measured counterclockwise, then

$$F_x = |\mathbf{F}| \cos \Theta \quad \text{and} \quad F_y = |\mathbf{F}| \sin \Theta \quad (2-9)$$

Note that the scalar components F_x and F_y take their sign from the angle Θ as the quadrant is established relative to the positive x -axis. Thus,

$$\mathbf{F} = F_x \mathbf{i} + F_y \mathbf{j} = F \cos \Theta \mathbf{i} + F \sin \Theta \mathbf{j} \quad (2-10)$$

and

$$\tan \Theta = \frac{F_y}{F_x}$$

Forces can be added in terms of their components. Consider two forces, $\mathbf{F}_1$ and $\mathbf{F}_2$, acting at a particular location as illustrated in figure 2-9. Determine the resultant force $\mathbf{F}_3$.

$$\begin{aligned} \mathbf{F}_3 &= \mathbf{F}_1 + \mathbf{F}_2 \\ &= F_{1x} \mathbf{i} + F_{1y} \mathbf{j} + F_{2x} \mathbf{i} + F_{2y} \mathbf{j} \\ &= (F_{1x} + F_{2x}) \mathbf{i} + (F_{1y} + F_{2y}) \mathbf{j} \end{aligned} \quad (2-11)$$

Example 2-1: If $\mathbf{F}_1$ equals 50.0 Newtons at 30° and $\mathbf{F}_2$ equals 30.0 Newtons at 130° as illustrated in figure 2-9, calculate the magnitude and direction of $\mathbf{F}_3$.

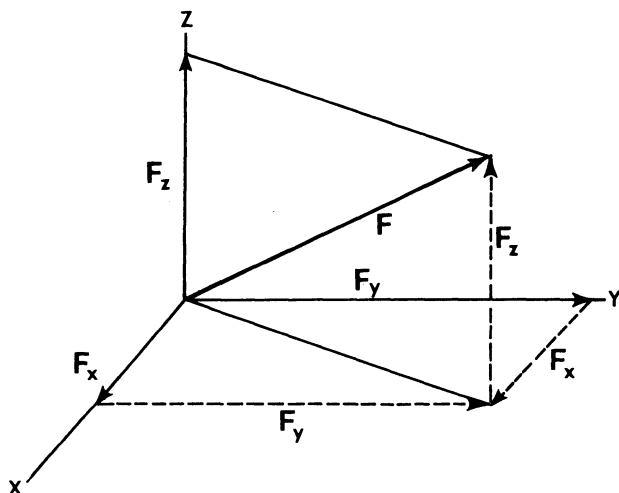


Figure 2-6. Resolving a vector into its components

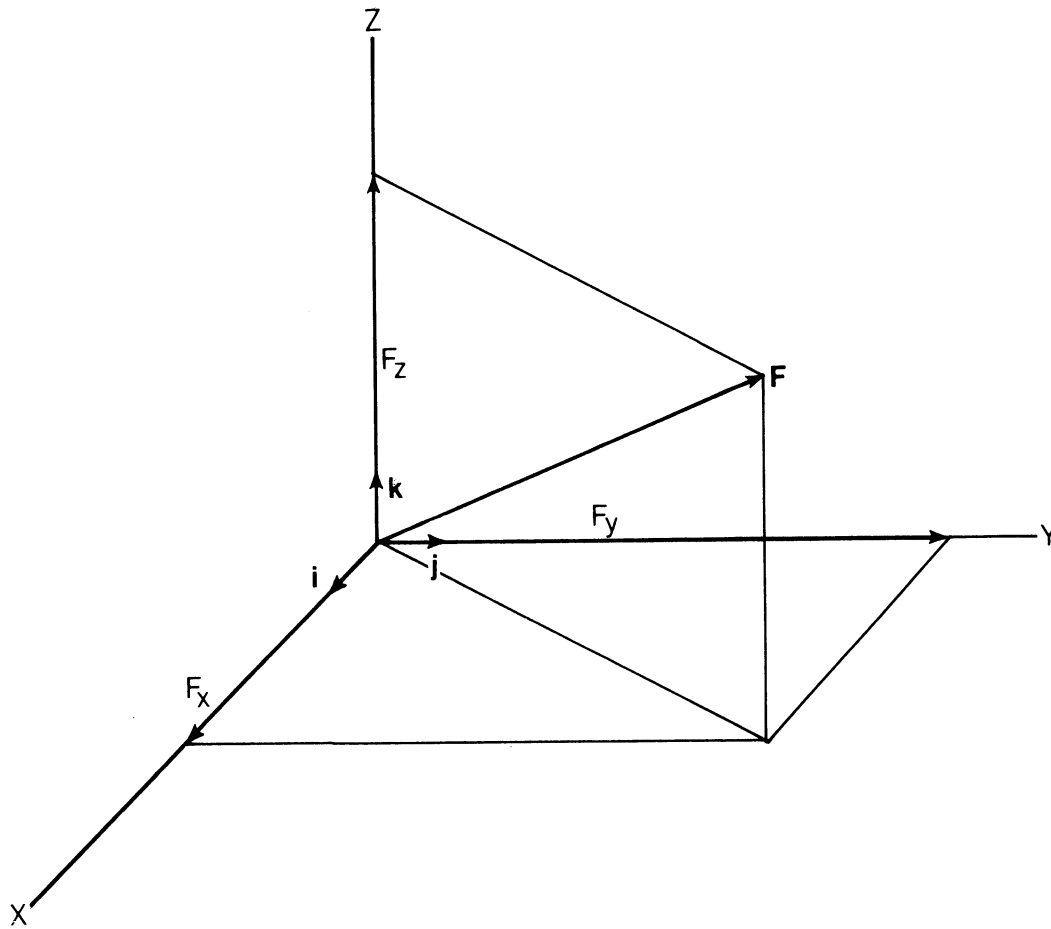


Figure 2-7. Concept of unit vectors

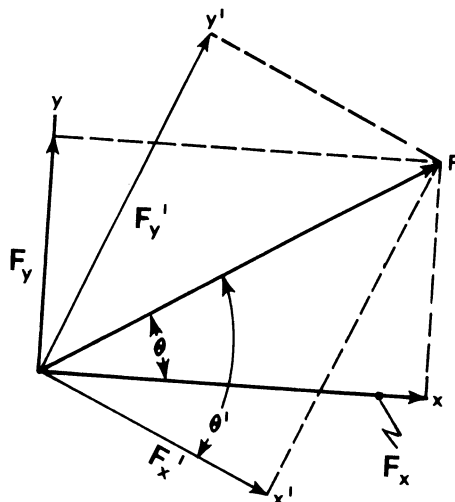


Figure 2-8. Change of axis orientation

Solution:

$$F_{1x} = 50 \cos 30^\circ = 43.3 \text{ N}$$

$$F_{2x} = 30 \cos 130^\circ = -19.3 \text{ N}$$

$$F_{1y} = 50 \sin 30^\circ = 25.0 \text{ N}$$

$$F_{2y} = 30 \sin 130^\circ = 23.0 \text{ N}$$

$$F_{3x} = 43.3 - 19.3 = 24.0 \text{ N}$$

$$F_{3y} = 25 + 23 = 48.0 \text{ N}$$

$$F_3 = \sqrt{24^2 + 48^2} = \boxed{53.7 \text{ Newtons}}$$

$$\tan \Theta_3 = \frac{F_{3y}}{F_{3x}} = \frac{48}{24}$$

$$\Theta_3 = \boxed{63.4^\circ}$$

2-4 Forces, Moments, and Couples

At this point in the development of basic equations governing problems in statics, a distinction should be

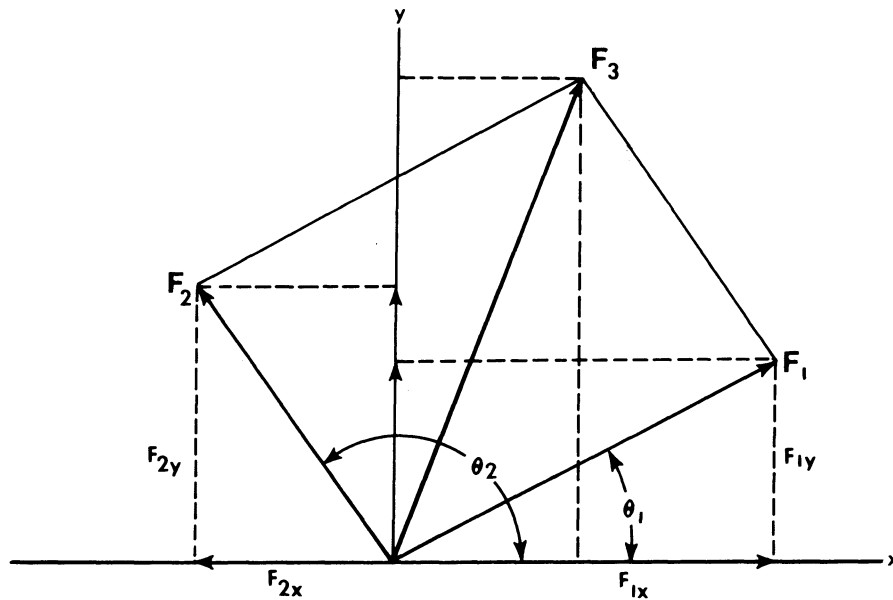


Figure 2-9. Addition of forces

made between the types of vectors used to represent various physical quantities.

Free vectors may be freely translated to any position parallel to the original line of action without affecting the analysis of the problem. Only the magnitude and direction must be preserved. Examples of free vectors are the displacement and velocity of a nonrotating rigid body and the concept of a couple as discussed in this section.

Sliding vectors may be positioned anywhere along the original line of action without affecting the analysis of the problem. Examples of sliding vectors are the forces that act on a rigid body and the tensile force in a “weightless” cable that supports some known weight. Moments caused by such vectors are not affected by changing the position of a vector along its line of action.

Fixed vectors are those whose effect depends on the local point of application. Examples of fixed vectors include forces acting on individual particles and forces acting on elastic or deformable bodies. The deformation of a rubber ball depends on the point of application of the force.

FORCE SYSTEMS

There are several special categories of force systems that should be distinguished.

Concurrent forces all have lines of action that pass through the same point. The analysis of forces on a particle or system of particles falls into this category, and therefore rotations caused by unbalanced moments are not considered in particle mechanics.

Colinear forces all have the same direction and the same line of action. This generally simplifies the analysis to that for a one-dimensional problem.

Coplanar forces all lie in the same plane, and therefore such force systems can be analyzed as two-dimensional problems.

MOMENTS

In the analysis of the effects of external and internal forces on rigid bodies, the concept of the moment of a vector about a line or axis must be introduced. Since the forces on a rigid body can be considered to be sliding vectors, the moment of such a force (torque) is defined as the product of the force times the perpendicular distance to the line of action of the force about some point in space.

Consider the force F acting on a rigid rod, both in the x - y plane as illustrated in figure 2-10. The rod is located with one end at the origin, and the point of application of the force is characterized by the position vector r . The moment of F about the z -axis in scalar form is

$$M = Fd = F(r \sin \Theta) \quad (2-12)$$

since $\sin \Theta$ equals $\sin (180 - \Theta)$. In vector form, the equation is

$$\mathbf{M} = \mathbf{r} \times \mathbf{F} \quad (2-13)$$

since equation 2-12 gives the magnitude of the vector product, called a *cross product*, as indicated in equation 2-13. It should also be noted that $F \sin \Theta$ represents

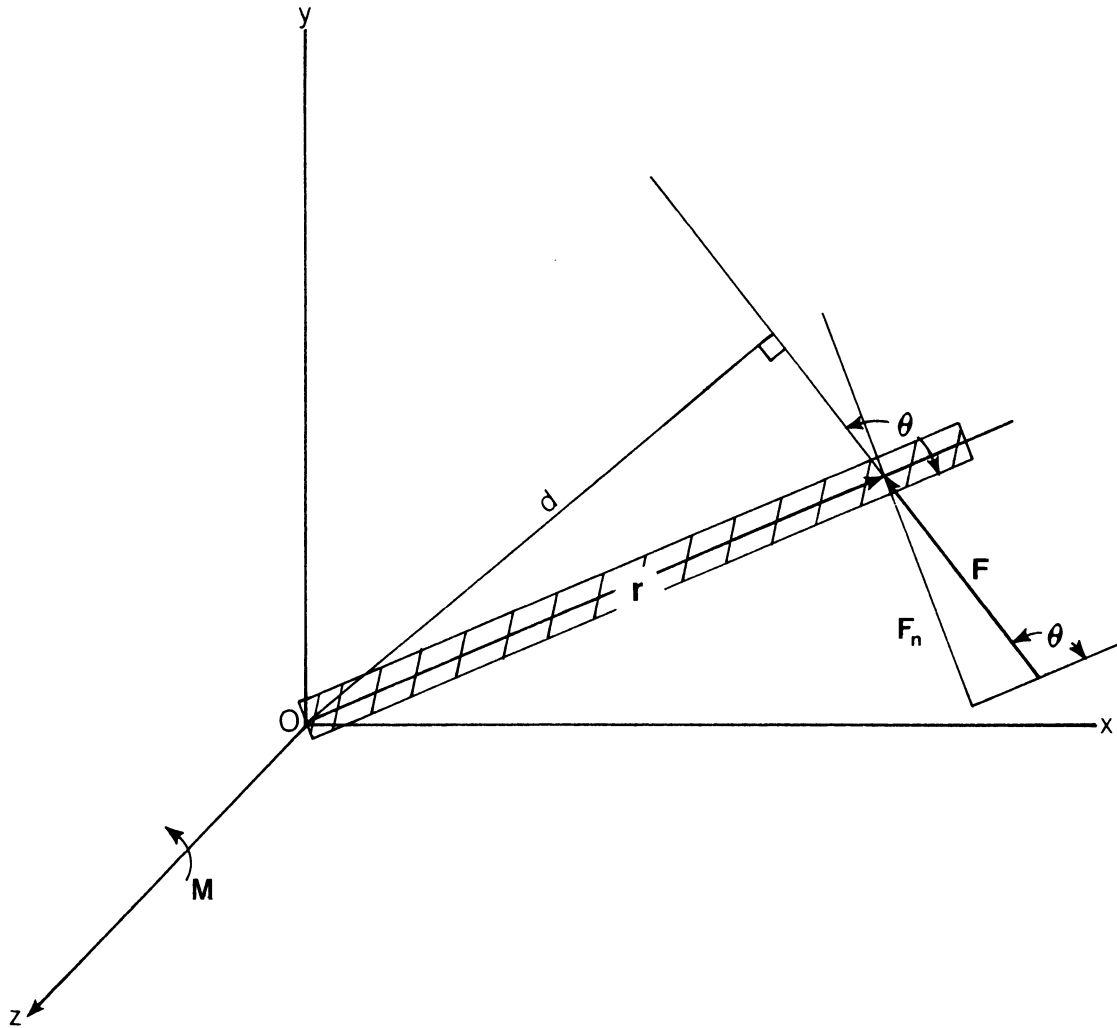


Figure 2-10. Moment generated by a force action on a rigid rod

the normal component of the force $\mathbf{F}$ with respect to the position vector $\mathbf{r}$, such that the magnitude of $\mathbf{M}$ can be found from

$$M = F_n r \quad (2-14)$$

where

$$F_n = F \sin \Theta$$

Thus, the moment of a vector about a line may also be found from the product of the distance to the point of application times the component of the force normal to the position vector $\mathbf{r}$.

The sign of the moment M is established by the *right-hand rule*, which is positive counterclockwise when Θ is between 0° and 180° .

If several forces are all applied at the same point, Varignon's theorem, or the "principle of moments," makes it possible to determine force

$$\mathbf{F}_R = \mathbf{F}_1 + \mathbf{F}_2 + \dots$$

and find the resultant moment

$$\mathbf{M}_R = \mathbf{r} \times \mathbf{F}_R \quad (2-15)$$

since

$$\begin{aligned} \mathbf{M}_R &= \mathbf{r} \times \mathbf{F}_1 + \mathbf{r} \times \mathbf{F}_2 + \dots \\ &= \mathbf{r} \times (\mathbf{F}_1 + \mathbf{F}_2 + \dots) \end{aligned}$$

This theorem can also be used to resolve the resultant force into x - and y -components and calculate the moment as indicated in figure 2-11.

$$M_R = -(r_x F_{Ry} + r_y F_{Rx})$$

where the minus sign results from $180^\circ < \Theta < 360^\circ$.

COUPLES

A *couple* is a pair of coplanar forces of equal magnitude acting in parallel but opposite directions, capable of causing rotation but not translation (i.e., a pure moment). Consider two equal and opposite forces, $\mathbf{F}_1$ and

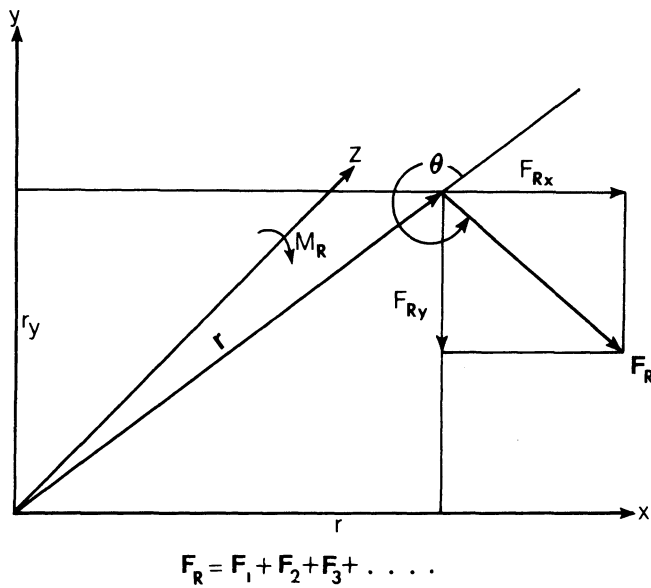


Figure 2-11. Moment of a resultant force

F_2 , acting in the x - y plane in figure 2-12. The moment of each force about the z -axis is

$$M_1 = r_1 \times F_1$$

$$M_1 = -r_1 \sin \theta F_1$$

$$M_2 = r_2 \times F_2$$

$$M_2 = r_2 \sin \theta F_2$$

The summation of moments about the origin is

$$M_0 = (r_2 \sin \theta - r_1 \sin \theta) F_1$$

since F_1 and $F_2 = F$

$$M_0 = (r_2 - r_1) \sin \theta F$$

$$= dF \quad (2-16)$$

This demonstrates that the moment of a couple is independent of the distance from the axis and is only a function of the magnitude of the forces and the perpendicular distance between the lines of action. Thus, the moment of a couple is a free vector that may be located anywhere in the plane of the couple.

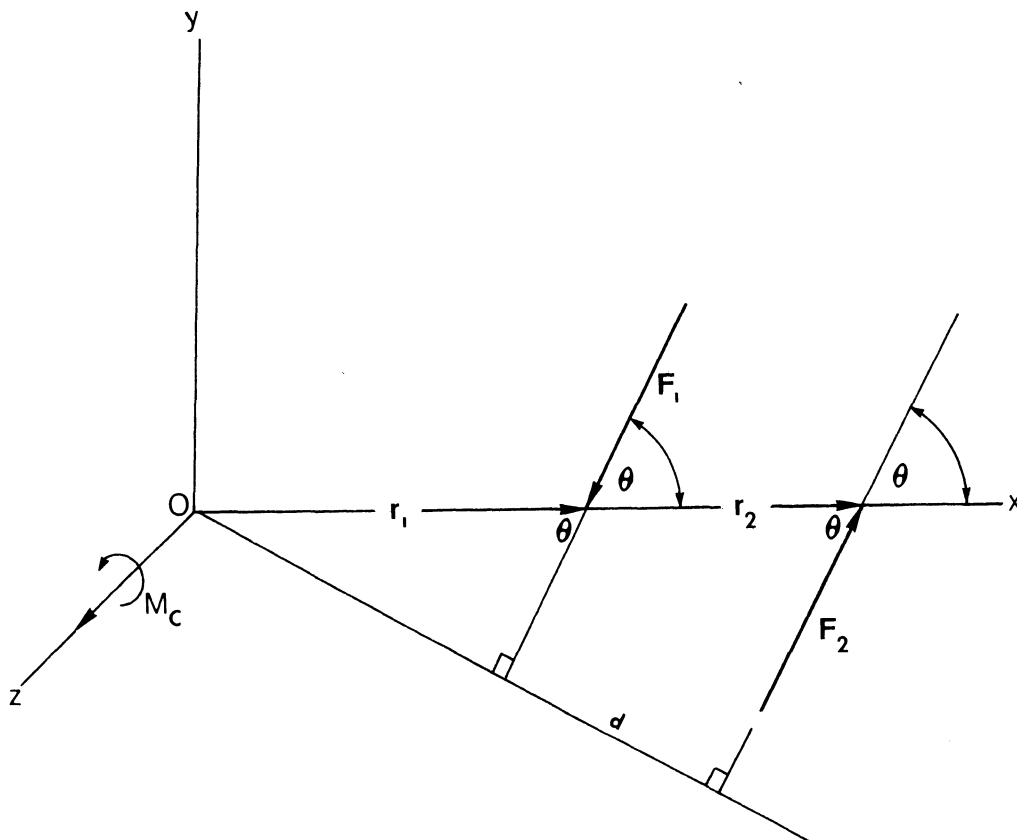


Figure 2-12. A couple

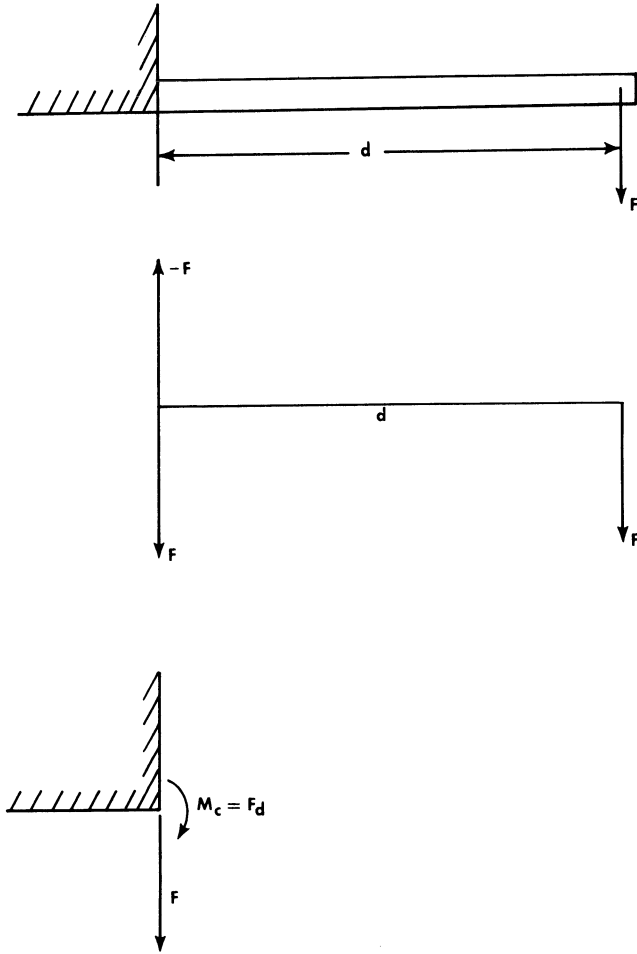


Figure 2-13. Replacement of a moment by a force and a couple

It is often useful to replace the moment of a force located at some distance from a particular location with a force acting at that location plus a couple about the axis passing through the location. Consider the force F acting on a rod attached rigidly to the wall, as in figure 2-13. The force acting on the wall may be found by replacing the force acting at perpendicular distance d from the wall with a force F acting at the wall and a couple moment $M_c = Fd$ also acting at the wall.

A system of forces acting at different locations on a rigid body may also be reduced to a single resultant force acting at the origin plus a couple that is the sum of all the couples and moments about the origin.

$$\mathbf{F}_R = \Sigma \mathbf{F} \quad (2-17)$$

$$\mathbf{M}_{R0} = \Sigma \mathbf{M}_0 \quad (2-18)$$

2-5 Static Equilibrium

The necessary and sufficient conditions for a rigid body to be in a state of static equilibrium are that the summation of forces and the summation of moments, as indicated in equations 2-17 and 2-18, be equal to zero.

$$\Sigma \mathbf{F} = 0 \quad (2-19)$$

$$\Sigma \mathbf{M}_0 = 0 \quad (2-20)$$

This will ensure that the body will experience neither translational (linear) motion nor rotational (angular) motion. Note that equations 2-19 and 2-20 provide a total of six equations that may be solved for six unknowns in a three-dimensional problem. Note also that if the forces are concurrent and are applied at the center of gravity of the body (taken as the origin), equation 2-20 is automatically satisfied, and the analysis becomes a problem involving the principles of particle mechanics (i.e., $\Sigma \mathbf{F}$ equals zero).

Most of the analyses in this text will involve coplanar force systems, however. In the case of real three-dimensional problems in ship stability, the analysis will be divided into longitudinal and transverse stability portions, so that coplanar force systems can be assumed in order to simplify the problem. For two-dimensional force systems, the equilibrium conditions reduce to a set of three scalar equations,

$$\Sigma F_x = 0 \quad (2-21a)$$

$$\Sigma F_y = 0 \quad (2-21b)$$

$$\Sigma M_0 = 0 \quad (2-21c)$$

which may be solved for a maximum of three unknowns. In certain types of problems, it is necessary to sum moments about more than one axis and eliminate one of the force summations. The resulting restrictions on linear independence of the equations are beyond the scope of this text, and the reader is referred to one of the standard texts concerning engineering statics (Beer and Johnston 1977; Hibbeler 1978).

FREE BODY DIAGRAMS

In order to apply the equations of equilibrium to a particular problem in statics, it is necessary to construct a *free body diagram*. This is done to account for all the forces and moments present, including the reaction forces, where the body comes in contact with another body, a supporting or restraining cable, or the ground plane. These reactions are generally classified in terms of the number of unknown forces and moments that may be present at the reaction.

1. One unknown force with a known line of action—

These reactions include cables, links, and situations where only a normal force to the surface can be supported. The latter includes assumed *frictionless surfaces* approximated by rollers, rockers, ball-bearing collars, and pins constrained in a particular direction such that the tangential force is always equal to zero (figure 2-14).

2. One unknown resultant force with an unknown line of action—

This type of reaction is found at pinned or hinged joints and rough surfaces that can support tangential as well as normal forces, but cannot support a couple or moment. If component forces are used,

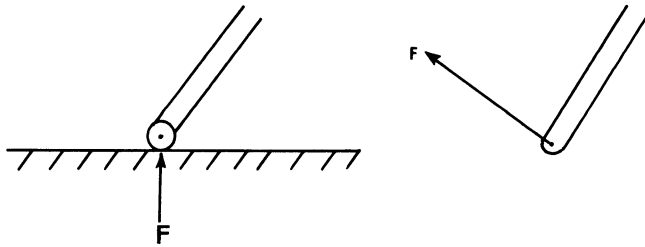


Figure 2-14. Frictionless support: one unknown force of known direction

there are still two unknown quantities involved (figure 2-15).

3. *One unknown resultant force with an unknown line of action plus an unknown couple*—This type of reaction is found at a fixed support that can support both forces and moments and can constrain the rigid body completely (figure 2-16).

Steps in the Construction of a Free Body Diagram

1. Sketch the contour or outline of the body detached (free) from other bodies or the ground. Dimension the sketch as required to calculate moments of the forces involved.

2. Indicate on the sketch all the external forces, both known and unknown, and the couples that may act on the body. This will include all reaction forces and couples as well as applied external loads and the weight of the body acting at the center of gravity.

3. Label the known forces and couples with the numerical values of their magnitude and direction in appropriate units for the required calculations. Label all the unknown forces and couples with letters appropriate to the equations of equilibrium. Remember that the sense of the direction of the external forces is determined by the effect of the surroundings *on the body*, not vice versa. The assumed direction for the unknown

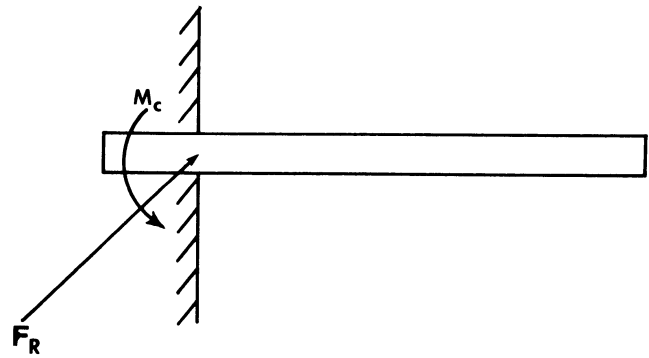


Figure 2-16. Fixed support: one unknown force of unknown direction and an unknown couple

forces is correct if the solution yields a positive number. A negative value for an unknown force indicates that the assumed direction of the force was incorrect.

The problem can now be solved, providing there are not more than three unknowns for two-dimensional problems involving the equations of static equilibrium.

Example 2-2: A rotating cargo crane on a merchant ship lifts a 10,000-lb load, as illustrated in figure 2-17(a). The boom is 60 feet long and will be considered weightless for purposes of the problem. The crane weighs 7,000 pounds, and its center of gravity can be assumed to be located 2 feet from the axis of rotation towards the boom (thus compensating for the weightless boom assumption). For the loading condition and angles indicated in the sketch, determine the compression load in the crane boom and the tension in the cable between A and B. Also determine the reaction forces and moment at the base of the crane (point D).

Solution: To determine the compression load in the crane boom $\overline{BC}$ and the cable tension in $\overline{AB}$, sketch a free body diagram of the boom, cable, and load, as illustrated in figure 2-17(b). Note that by neglecting the

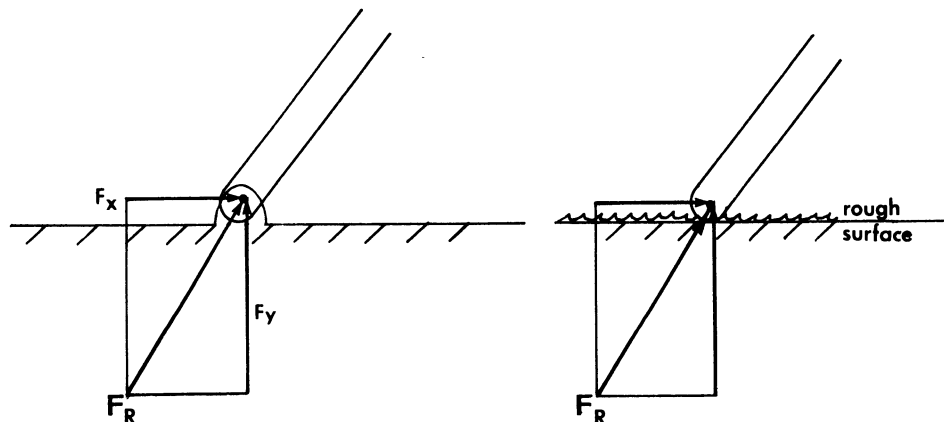


Figure 2-15. Pinned support: one unknown force of unknown direction

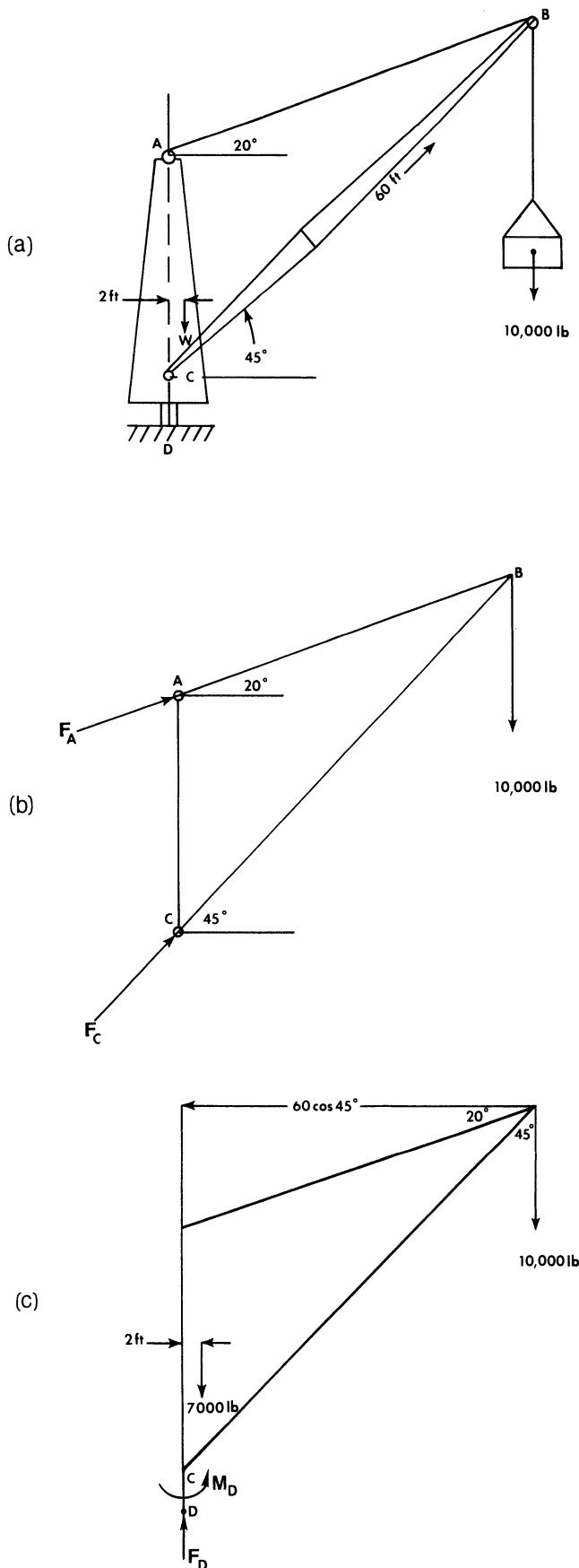


Figure 2-17(a),(b),(c). Cargo crane

weight of the boom, the remaining forces are concurrent through point B. Thus, the condition that ΣF equals zero at point B is sufficient to determine the unknown reaction forces F_A and F_C .

$$\Sigma F_x = 0$$

$$F_A = \cos 20^\circ + F_C \cos 45 = 0$$

$$\Sigma F_y = 0$$

$$F_A \sin 20^\circ + F_C \sin 45 - 10,000 = 0$$

$$F_A = -F_C \frac{\cos 45}{\cos 20} \text{ from } \Sigma F_x = 0$$

$$-F_C \cos 45 \frac{\sin 20}{\cos 20} + F_C \sin 45 = 10,000$$

$$F_C (\sin 45 - \cos 45 \tan 20) = 10,000 \text{ lb}$$

$$F_C = \frac{10,000}{0.45} = \boxed{22,235 \text{ lb}}$$

$$F_A = -F_C \frac{\cos 45}{\cos 20} = \boxed{-16,730 \text{ lb}}$$

Thus, the assumed direction of F_A on the free body diagram was incorrect; i.e., the cable is in tension.

To determine the reaction at the base of the crane, point D, sketch a free body diagram of the entire crane as illustrated in figure 2-17(c).

By inspection,

$$\Sigma F_x = 0$$

$$\Sigma F_y = 0$$

$$F_D - 7,000 \text{ lb} - 10,000 \text{ lb} = 0$$

$$F_D = \boxed{17,000 \text{ lb}}$$

$$M_0 = 0 \text{ about } D$$

$$M_D - 7,000 (2) - 10,000 (60 \cos 45) = 0$$

$$M_D = 14,000 + 424,300$$

$$= \boxed{438,300 \text{ ft-lb}}$$

2-6 Analysis of Distributed Quantities

In the preceding discussion, the weight of an object was assumed to be concentrated at a single point so that the force and moment equations would be easy to apply. For simple geometrical shapes, such as rectangular or circular bars, the position of the center of gravity (where the sum of all the gravitational forces is assumed to be concentrated) is quite easy to calculate. For more complex shapes such as ships, integrals and more complex

summations must be used. For situations where the weight (density) of the component parts is uniform or can be described by an equation, an integral formulation of the properties of distributed quantities can be made. When the component weights have discrete values, a numerical summation must be used. When using a digital computer for the calculations, numerical integration methods are used, which are discussed in appendix B.

THE FIRST MOMENT OF AREAS AND VOLUMES: THE CENTROID

The center of gravity or centroid of an object with a uniform weight per unit area or volume may be found by using the first moment of area or volume. Since the weight density is uniform, it may be eliminated from the calculation, making the problem one of geometric analysis.

Two-dimensional Shapes

Consider a flat horizontal plate of homogeneous density and thickness aligned with the x - y plane, as in figure 2-18. The centroid of an arbitrary section of the plate may be found by using the first moment of area about the x - and y -axes. The first moment of area about the x -axis is defined as

$$M_x = \int y dA \quad (2-22)$$

And the first moment of area about the y -axis is defined as

$$M_y = \int x dA \quad (2-23)$$

Plane shapes having at least one side coinciding with an axis are frequently found in naval architecture. These may include situations where the ordinate values are known functions of the abscissa, i.e., y equals $f(x)$ at discrete values of the abscissa and at spacings δx_i . For

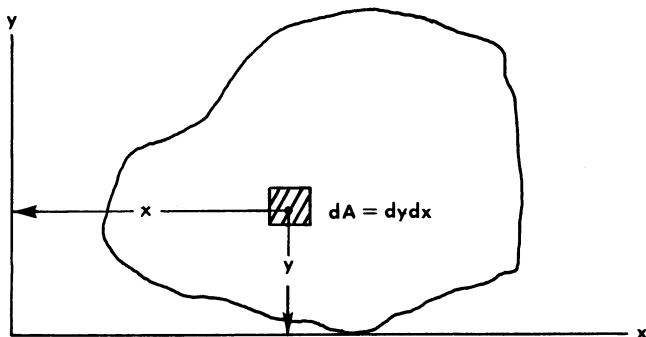


Figure 2-18

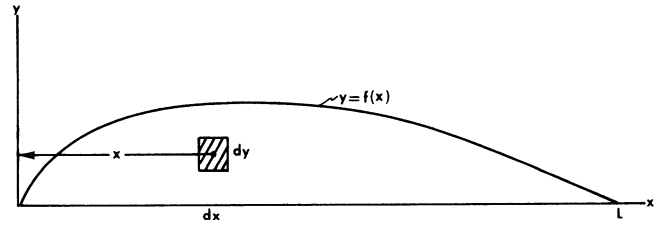


Figure 2-19

this situation, illustrated in figure 2-19,

$$\begin{aligned} M_x &= \int_0^L \int_0^y y \, dy \, dx \\ M_x &= \int_0^L \frac{y^2}{2} \, dx \end{aligned} \quad (2-24)$$

which may be approximated numerically by using N rectangles of height y and width δx_i

where

$$i = 1, 2, \dots, N$$

$$M_x = \sum_{i=1}^N \frac{y^2}{2} \delta x_i \quad (2-24a)$$

Similarly the first moment of area about the y -axis is

$$\begin{aligned} M_y &= \int_0^L \int_0^y x \, dy \, dx \\ M_y &= \int_0^L xy \, dx \end{aligned} \quad (2-25)$$

which may be approximated numerically by

$$M_y = \sum_{i=1}^N xy \delta x_i$$

The centroid of the plane shapes discussed above may be found from

$$\bar{x} = \frac{M_y}{A} = \frac{\int x dA}{\int dA} \quad (2-26)$$

$$\bar{y} = \frac{M_x}{A} = \frac{\int y dA}{\int dA} \quad (2-27)$$

where A = total area.

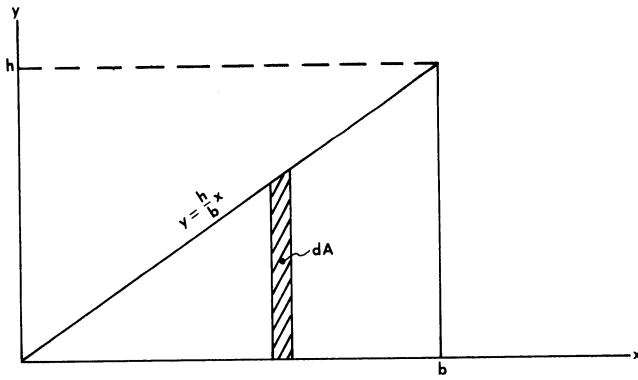


Figure 2-20

Example 2-3: Compute the centroid of the right triangle shown in figure 2-20.

Solution: Since the shape of the triangle may be expressed in equation form, and the base of the triangle corresponds to the x -axis, the integral method using equations 2-24 and 2-25 may be used.

$$M_x = \int_0^b \frac{y^2}{2} dx$$

$$= \int_0^b \frac{1}{2} \left(\frac{h}{b} x \right)^2 dx = \frac{1}{2} \frac{h^2}{b^2} \frac{x^3}{3} \Big|_0^b$$

$$M_x = \frac{bh^2}{6}$$

$$M_y = \int_0^b xy \, dx$$

$$= \int_0^b x \frac{h}{b} x \, dx = \frac{h}{b} \frac{x^3}{3} \Big|_0^b$$

$$M_y = \frac{b^2h}{3}$$

Thus,

$$\bar{x} = \frac{M_y}{A} = \frac{b^2h}{3} \Big/ \frac{bh}{2} = \frac{2}{3}b$$

$$\bar{y} = \frac{M_x}{A} = \frac{bh^2}{6} \Big/ \frac{bh}{2} = \frac{h}{3}$$

as expected.

Composite Shapes

In many instances, the centroid of two-dimensional shapes consisting of rectangles, triangles, and other common shapes may be calculated by summing the moments

about a common axis for the individual components and setting the sum equal to the total area of the shape multiplied by the centroid of the composite area.

$$M_x = \sum_{i=1}^N y_i A_i$$

$$M_y = \sum_{i=1}^N x_i A_i$$

$$\bar{x} = \frac{M_y}{A} = \frac{\sum x_i A_i}{\sum A_i} \quad (2-28)$$

$$\bar{y} = \frac{M_x}{A} = \frac{\sum y_i A_i}{\sum A_i} \quad (2-29)$$

where N = number of shapes.

Example 2-4: Locate the centroid of the V-block shape shown in figure 2-21(a).

Solution: The origin of the coordinate system may be placed anywhere on or near the section and would normally be chosen on a line of symmetry to simplify the calculation. In order to demonstrate the method of composite sections, however, the origin will be chosen at the lower left-hand side of the shape, as illustrated in figure 2-21(b). The section is then divided into one

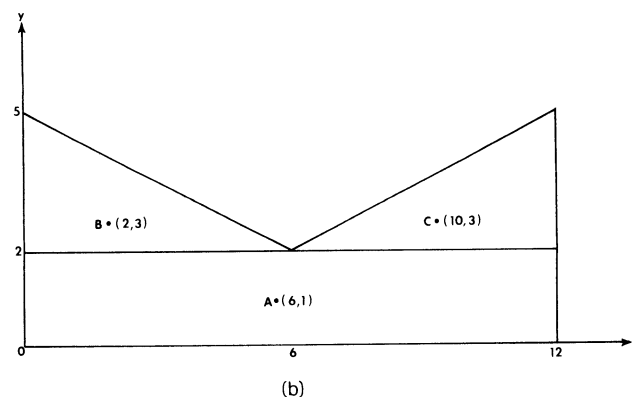
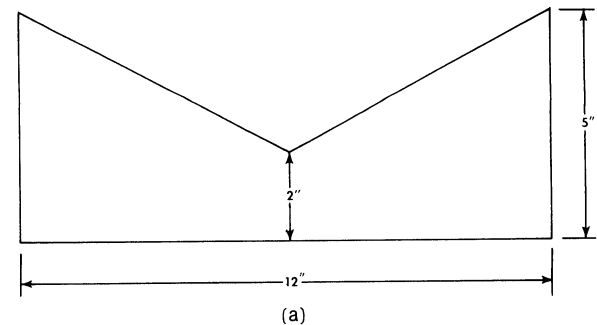


Figure 2-21(a),(b). V-block

rectangular area A and two triangular areas B and C , with centroids as indicated. Thus,

$$A_A = 2 \times 12 = 24 \text{ in}^2$$

$$A_B = A_C = \frac{6 \times 3}{2} = 9 \text{ in}^2$$

$$\bar{x} = \frac{\sum x_i A_i}{\sum A_i} = \frac{6(24) + 2(9) + 10(9)}{24 + 9 + 9}$$

$$= \frac{252}{42} = \boxed{6 \text{ in}} \text{ as expected}$$

$$\bar{y} = \frac{\sum y_i A_i}{\sum A_i} = \frac{1(24) + 3(9) + 3(9)}{24 + 9 + 9}$$

$$= \frac{78}{42} = \boxed{1.86 \text{ in}}$$

An alternate solution for $\bar{y}$ could involve subtracting the V-notch area from the enclosing rectangle.

$$\bar{y} = \frac{2.5(5 \times 12) - 4\left(\frac{3 \times 12}{2}\right)}{5 \times 12 - \frac{3 \times 12}{2}} = \frac{78}{42} = \boxed{1.86 \text{ in}} \text{ as before.}$$

The centroids of other standard shapes are given in handbooks and in textbooks on statics and strength of materials.

Three-dimensional shapes. The centroid of a three-dimensional shape may be found from the first moment of volume about the three principal axes. Thus,

$$\bar{x} = \frac{\int x dV}{V} = \frac{\sum_{i=1}^N \bar{x}_{c_i} A_{yzi} \delta x_i}{\sum_{i=1}^N A_{yzi} \delta x_i} \quad (2-30a)$$

$$\bar{y} = \frac{\int y dV}{V} = \frac{\sum_{i=1}^N \bar{y}_{c_i} (A_{xz} \delta y)_i}{\sum_{i=1}^N (A_{xz} \delta y)_i} \quad (2-30b)$$

$$\bar{z} = \frac{\int z dV}{V} = \frac{\sum_{i=1}^N \bar{z}_{c_i} (A_{xy} \delta z)_i}{\sum_{i=1}^N (A_{xy} \delta z)_i} \quad (2-30c)$$

where $\bar{x}_{c_i}$ = centroid of individual components

A_{yzi} = component area in a yz plane

δx_i = thickness of layer A_{yzi}

N = number of components.

Numerical integration techniques for making these calculations are found in appendix B.

For composite three-dimensional shapes consisting of known component weights W_i and corresponding centers of gravity $\bar{x}_i$, $\bar{y}_i$, and $\bar{z}_i$, of this inhomogeneous combination, the centroid may be found by summing moments of the weights or masses about the three principal axes.

$$\bar{x} = \frac{\sum \bar{x}_i W_i}{\sum W_i} \quad (2-31a)$$

$$\bar{y} = \frac{\sum \bar{y}_i W_i}{\sum W_i} \quad (2-31b)$$

$$\bar{z} = \frac{\sum \bar{z}_i W_i}{\sum W_i} \quad (2-31c)$$

This formulation will be used in chapter 6 to determine the changes in the center of gravity of a ship.

2-7 Second Moment of Area: Moment of Inertia

The second moment of area, which derives its name from the power on the variable in the defining equation, is generally called the *moment of inertia*, regardless of whether or not the equation includes the mass density that relates the quantity to “inertia.” It is defined with respect to a given axis as follows:

$$I_x = \int y^2 dA \quad (2-32)$$

$$I_y = \int x^2 dA \quad (2-33)$$

The moment of inertia is always a positive quantity and is used in the basic equations for strength of materials (chapter 5), ship hydrostatics (chapters 6 and 7), ship motions (chapter 11), and in section 2-9 of this chapter. Moments of inertia are tabulated for many structural shapes in handbooks on civil and mechanical engineering. Statics and strength of materials textbooks also contain tables of moments of inertia for standard shapes (Beer and Johnston 1977; Hibbeler 1978). Unfortunately, many ship applications involve calculating the moment of inertia of the “waterplane area” for various cross sections of the ship, and these are rarely standard shapes. However, they do have one side coinciding with an axis when calculations about the cen-

terline are involved. Thus, the equations can be set up for numerical integration in a fashion similar to that for the centroid calculation illustrated in figure 2-17 of the previous section.

$$I_x = \int_0^L \int_0^y y^2 dy dx = \int_0^L \frac{y^3}{3} dx \quad (2-34)$$

$$= \sum_{i=1}^N \frac{y_i^3}{3} \delta x_i$$

$$I_y = \int_0^L \int_0^y x^2 dy dx = \int_0^L x^2 y dx \quad (2-35)$$

$$= \sum_{i=1}^N x_i^2 y_i \delta x_i$$

Since most ships are symmetrical about the centerline, hull-form offsets (y -values for each x_i) are based on one-half of a ship or "half breadth," as explained in chapter 3. Therefore, the moment of inertia of the whole ship about the centerline is two times the moment of inertia for one side about the centerline.

Example 2-5: Calculate the moment of inertia of the triangle in example 2-3 about the x -axis.

Solution:

$$\begin{aligned} I_x &= \int y^2 dA = \int_0^b \frac{y^3}{3} dx \\ &= \frac{1}{3} \int_0^b \left(\frac{hx}{b} \right)^3 dx \\ &= \frac{1}{3} \left(\frac{h}{b} \right)^3 \frac{x^4}{4} \Big|_0^b \\ &= \boxed{\frac{bh^3}{12}} \end{aligned}$$

This can be shown to be true for all triangles about the base.

PARALLEL AXIS THEOREM

It is frequently necessary to calculate the moment of inertia of an area about some other axis parallel to the axis through the centroid of the area for which the moment of inertia is known. This will be required in the calculation of the moment of inertia of composite areas. The theorem can be developed by calculating the moment of inertia of an arbitrary shape about the x -axis using y -values related to the distance d from the axis passing through the centroid c to the new axis, as illustrated in figure 2-22.

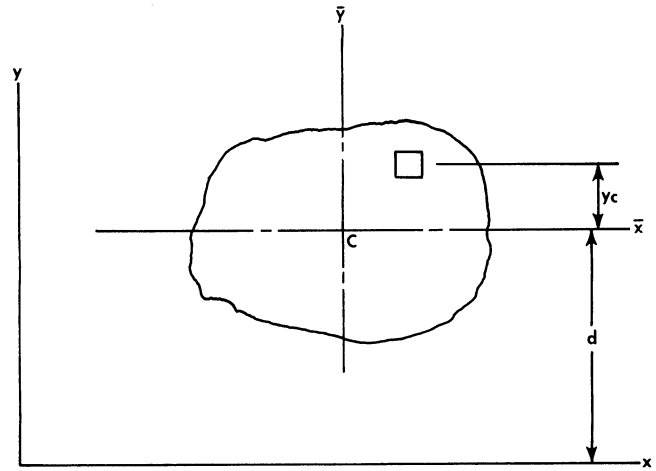


Figure 2-22. Parallel axis theorem

$$\begin{aligned} I_x &= \int y^2 dA \quad y = y_c + d \\ &= \int (y_c + d)^2 dA \\ &= \int y_c^2 dA + d \int y_c dA + d^2 \int dA \\ &= I_x + dM_c + d^2 A \end{aligned}$$

where I_x = moment of inertia about the centroid

d = parallel distance the axis is shifted

M_c = moment of area about the centroid

= 0 by definition

y_c = distance of elemental area from the centroidal axis.

Thus,

$$I_x = I_x + d^2 A \quad (2-36)$$

Similarly, letting d equal the distance from the y axis to the vertical centroidal axis,

$$I_y = I_y + d^2 A \quad (2-37)$$

Example 2-6: Determine the moment of inertia of a rectangle with respect to an axis parallel to its base and passing (a) through its centroid, and (b) about the base, as in figure 2-23.

Solution: Using equation 2-32 and noticing the symmetry about the x -axis passing through the centroid,

$$\begin{aligned} I_x &= 2 \int_0^b \int_0^{h/2} y^2 dy dx \\ &= 2 \frac{y^3}{3} \Big|_0^{h/2} x \Big|_0^b = \frac{2}{3} \frac{h^3}{8} b \\ &= \boxed{\frac{bh^3}{12}} \end{aligned}$$

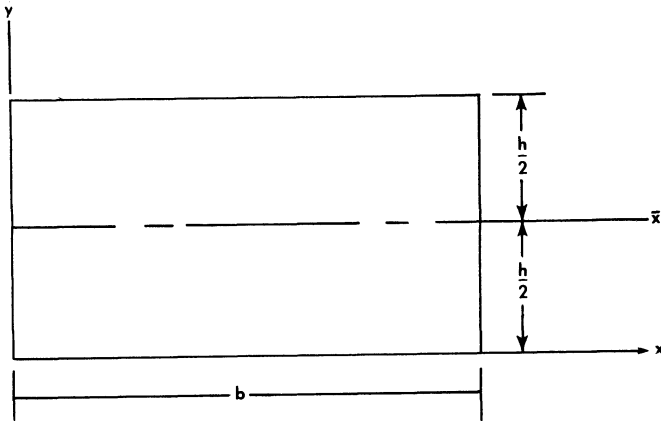


Figure 2-23

To calculate the moment of inertia about the base, the parallel axis theorem can be used.

$$\begin{aligned} I_x &= I_{\bar{x}} + d^2 A \\ &= \frac{bh^3}{12} + \left(\frac{h}{2}\right)^2 bh \\ &= \boxed{\frac{bh^3}{3}} \end{aligned}$$

As a check, return to equation 2-32 and integrate about the base.

$$\begin{aligned} I_x &= \int_0^b \int_0^h y^2 dy dx \\ &= \frac{y^3}{3} \bigg|_0^h x \bigg|_0^b \\ &= \frac{bh^3}{3} \text{ as before.} \end{aligned}$$

Moment of Inertia of Composite Areas

The moment of inertia of composite areas consisting of combinations of simple shapes may be calculated by adding or subtracting the moments of inertia of the shapes making up the composite area, using the parallel axis theorem for any shapes whose centroids are located off the centroidal axis of the composite area.

Example 2-7: Calculate the moment of inertia of an I-beam that is made from three sections of 2"-by-8" steel plate welded together as illustrated in figure 2-24.

Solution: Both an additive method that requires the use of the parallel axis theorem and a subtractive method will be used for comparison.

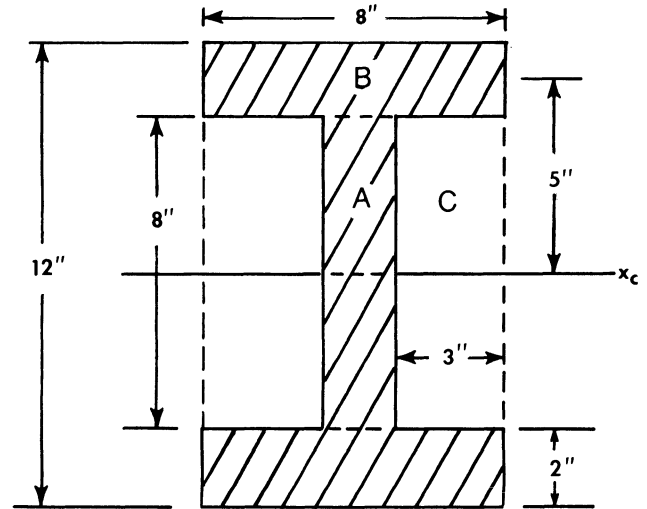


Figure 2-24. I-beam

Additive method:

$$\begin{aligned} I_x &= I_A + 2[I_B + d^2 A_B] \\ &= \frac{2(8)^3}{12} + 2 \left[\frac{8(2)^3}{12} + 5^2(8 \times 2) \right] \\ &= \boxed{896 \text{ in}^4} \end{aligned}$$

Subtractive method:

$$\begin{aligned} I_x &= I_{8 \times 12} - 2I_c \\ &= \frac{8(12)^3}{12} - 2 \left(\frac{3(8)^3}{12} \right) \\ &= \boxed{896 \text{ in}^4} \end{aligned}$$

RADIUS OF GYRATION

Occasionally it is desirable to relate a plane area to its moment of inertia by defining a radius of gyration k_x or k_y .

$$I_x = k_x^2 A \quad \text{or} \quad k_x = \sqrt{\frac{I_x}{A}} \quad (2-38)$$

$$I_y = k_y^2 A \quad \text{or} \quad k_y = \sqrt{\frac{I_y}{A}} \quad (2-39)$$

This is equivalent to concentrating the area into a thin strip of negligible inertia about its own centroidal axis and locating it at a distance from the appropriate axis equal to the radius of gyration (see parallel axis theorem). This concept is used extensively in structural mechanics, dynamics, and ship-motion studies.

COEFFICIENT OF INERTIA: C_I

In ship design it is convenient to relate the moment of inertia of a waterplane area (the area of this intersection

of the ship's hull at a particular draft with the surface of the ocean) with that of a circumscribing rectangle having the same waterline length and beam as the ship.

$$I_y = I_L = C_{IL}BL^3/12 \quad (2-40a)$$

$$I_x = I_T = C_{IT}LB^3/12 \quad (2-40b)$$

where C_{IL} = longitudinal waterplane inertia coefficient

C_{IT} = transverse waterplane inertia coefficient.

This concept will be discussed further in chapters 3 and 6.

2-8 Hydrostatics

FLUID PROPERTIES

Hydrostatics is the study of liquids at rest. Liquids and gases are fluids (i.e., substances that cannot sustain shear forces when in static equilibrium and that deform continuously when subjected to shear stresses while in motion). The following definitions are needed to describe the basic hydrostatic properties of fluids:

1. *Density and specific weight*—The density ρ of a fluid is its mass per unit volume, while the specific weight w is its weight per unit volume. The system of units relating the two concepts can be obtained from Newton's second law applied to the acceleration caused by gravity g ; i.e.,

$$F = ma \quad \text{becomes} \quad W = mg \quad (2-41)$$

where W = weight of body of mass m .

Thus, in the English gravitational system of units, an lbf is equivalent to a slug-ft/sec², and in the metric system, a Newton is equivalent to a kilogram-meter/sec². If both sides of the expression are divided by the volume of the substance, one obtains

$$\frac{W}{V} = \frac{m}{V} g$$

Thus the specific weight w and the density ρ are related by

$$w = \rho g \quad (2-42)$$

2. *Specific volume*—The specific volume of a fluid is its volume per unit weight, which is the reciprocal of the specific weight. Since w will also be used to indicate weight changes on ships, ρg will generally be used to indicate specific weight. Thus,

$$v = \frac{1}{w} = \frac{1}{\rho g} \quad (2-43)$$

3. *Specific gravity*—The specific gravity γ of a liquid is the ratio of its density to that of pure water at a

standard temperature. Physicists use 39.2°F (4°C) as the standard, but engineers and naval architects ordinarily use 59°F (15°C). In the metric system, the density of water at 4°C is 1.00 g/cm³, hence the specific gravity (which is dimensionless) has the same numerical value for a liquid as its density in that system. Since the density of a fluid varies with temperature, specific gravities must be determined and specified at particular temperatures as shown in appendix A.

For purposes of making approximate engineering calculations, the following table gives commonly accepted values of the properties of fresh water and salt water in both English and metric units. These values should be used in solving homework problems unless otherwise specified or implied by some specific temperature being given.

	<i>English</i>	<i>Metric</i>
<i>Specific weight of fresh water</i>	62.4 lbf/ft ³	.0098MN/m ³
<i>Specific weight of salt water</i>	64.0 lbf/ft ³	.01MN/m ³
<i>Specific volume of fresh water</i>	36 ft ³ /ton	102 m ³ /MN
<i>Specific volume of salt water</i>	35 ft ³ /ton	99.5 m ³ /MN
<i>Mass density of fresh water</i>	1.94 slugs/ft ³	1,000 kg/m ³
<i>Mass density of salt water</i>	1.99 slugs/ft ³	1,025 kg/m ³
<i>Specific gravity of salt water</i>	1.025	1.025
<i>Acceleration of gravity</i>	32.17 ft/sec ²	9.807 m/sec ²

Note that 1,000 kg equals a metric ton in mass units and that MN stands for meganewton (10⁶ Newtons) in force units. The English long ton (2,240 lbf) is equivalent to 1.01605 metric tons.

The specific weight of a liquid varies only slightly with pressure; it also depends on temperature, and the variation may be considerable. Since specific weight w is equal to ρg , it depends on the local value of the acceleration of gravity in addition to the variations with temperature and pressure. The text deals specifically with ships at sea level, however, where g can be considered constant, so this variation can be neglected. The variation of the specific weight of water with temperature under normal conditions, where g equals 32.17 fps/sec, is shown in appendix A.

PRESSURE

There are no shear stresses in fluids at rest; hence, only normal stresses are present. The average pressure intensity is defined as the normal force exerted on a unit area. If F represents the normal force on some finite

area A , while dF represents the force on an infinitesimal area dA , the pressure is defined as follows:

$$p = \lim_{\delta A \rightarrow 0} \frac{\delta F}{\delta A} = \frac{dF}{dA} \quad (2-44)$$

If the pressure is uniform over the total area, then p equals F/A . In engineering units, pressure is generally expressed in pounds per square inch (psi) or pounds per square foot (psf). In a solid, because of the possibility of tangential stresses between adjacent particles, the stresses at a given point may be different in different directions. But in a fluid at rest, no tangential stresses can exist, and the only forces between adjacent surfaces are pressure forces normal to the surfaces. Therefore, the pressure at any point in a fluid at rest is the same in every direction and is a scalar quantity.

PRESSURE-DEPTH RELATIONSHIP

The pressure in a fluid at rest changes only in the direction of gravitational acceleration. Thus, the pressure in a horizontal plane is everywhere the same for fluids of the same density. To determine the variation of pressure in a vertical direction, consider a small volume of fluid at rest, as in figure 2-25. The weight of the fluid in the volume equals $\rho g A \delta z$. Since the summation of forces equals zero,

$$\begin{aligned} (p + \delta p)A - pA - \rho g A \delta z &= 0 \\ \delta p &= \rho g \delta z \\ \text{or } \rho g &= \lim_{\delta z \rightarrow 0} \frac{\delta p}{\delta z} = \frac{dp}{dz} \end{aligned} \quad (2-45)$$

If the fluid can be considered incompressible (i.e., the specific weight is nearly constant for the range of height or depth variation in a particular problem), the derivative form can be replaced by the integration of equa-

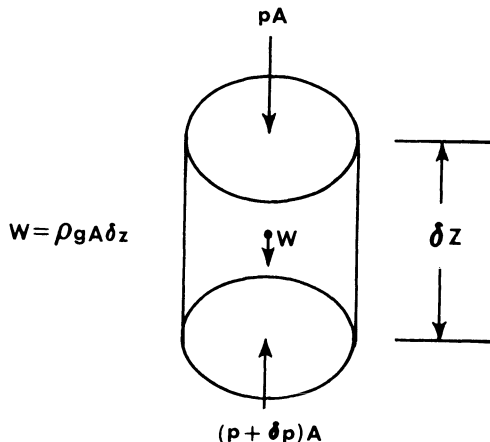


Figure 2-25

tion 2-45, and one obtains

$$\delta p = \rho g h \quad (2-46)$$

where h = change in depth.

ABSOLUTE AND GAGE PRESSURE

A relationship for calculating the change in pressure with depth was obtained in the previous section. A reference pressure is thus required to establish the value of the pressure at any point. Pressures are generally measured relative to zero normal stress (absolute pressure) or to the zero reading on a pressure gage (gage pressure). The relationship between these two reference pressures is illustrated in figure 2-26, where it is assumed that atmospheric pressure exists on the exterior of the gage-pressure measuring device. This relationship is summarized by the following equation:

$$P_{\text{abs}} = P_{\text{atm}} + P_{\text{gage}} \quad (2-47)$$

where P_{abs} = absolute pressure

P_{atm} = atmospheric pressure

P_{gage} = gage pressure.

The variation of pressure in a liquid at rest is thus:

$$p = p_0 + \rho g h \quad (2-48)$$

where p_0 is the reference pressure at the liquid surface. If the reference pressure is taken to be zero gage pressure, then the stated pressure p is a gage pressure. If the reference pressure is taken to be the atmospheric pressure, then the pressure will be an absolute pressure. The standard atmospheric pressure is taken to be 14.7 psi, which is equivalent to 29.92 inches of mercury (760 mm of mercury) with a specific gravity of 13.6.

2-9 Hydrostatic Forces on Submerged Surfaces

As can be seen from equation 2-48, the pressure varies linearly with changes in depth, provided that gage pressure is used and that the changes in depth do not involve changes in the density of water. When a fluid is at rest no tangential forces can be sustained, so the hydrostatic forces are always normal to the surface. Consider the following situations:

1. Forces on a horizontal plane surface.
 - a) The magnitude of the force is equal to the integral of the pressure over the area.

$$\begin{aligned} F &= \int p dA = \int \rho g h dA = \rho g h \int dA \\ &= \rho g h A \end{aligned} \quad (2-49)$$

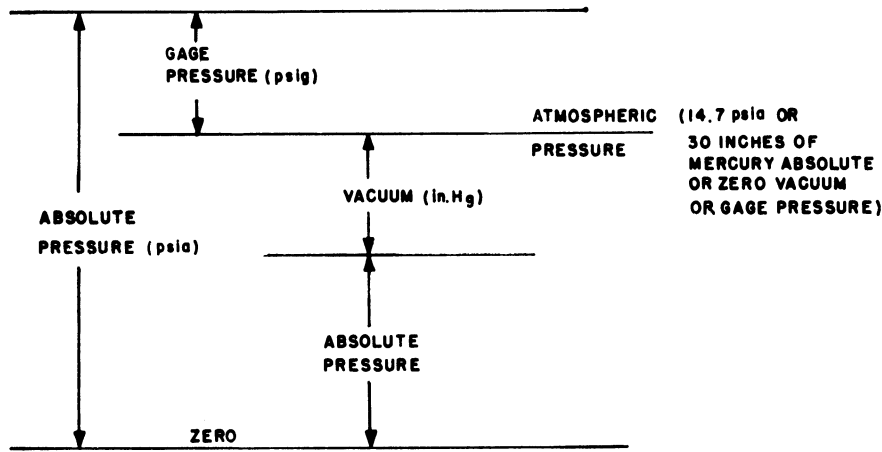


Figure 2-26. Relationships among gage pressure, atmospheric pressure, vacuum, and absolute pressure

- b) The point of application of the force is at the centroid of the horizontal area.
2. Forces on an inclined plane surface.
- a) The magnitude of the force is equal to the integral of the pressure over the area such as AB illustrated in figure 2-27.

$$F = \int_A^B p dA = \rho g \int_A^B h dA$$

Now

$$h = y \sin \theta$$

$$F = \rho g \sin \theta \int_A^B y dA$$

but

$$\int_A^B y dA = \bar{y}A = \frac{\bar{h}A}{\sin \theta}$$

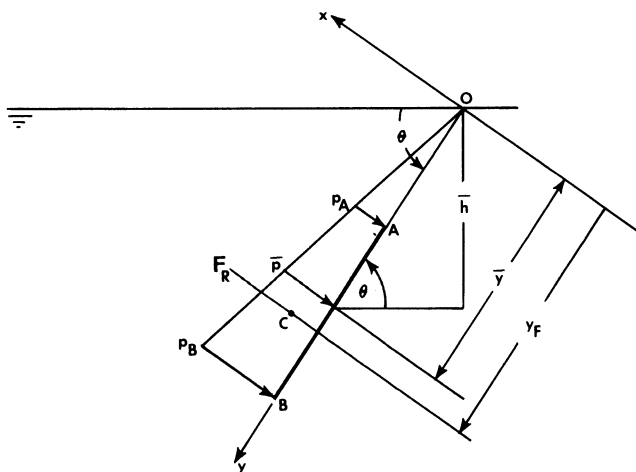


Figure 2-27. Force on an inclined surface

Thus,

$$F = \rho g \bar{h}A = \bar{p}A \quad (2-50)$$

where $\bar{h}$ = depth of the centroid of the area
 $\bar{p}$ = mean pressure at the centroid of the area.

- b) The point of application of the resultant force is such that the moment of the resultant force about a horizontal axis at the surface is equal to the moment of the distributed force about the same axis.

$$\begin{aligned} y_F F_R &= \int y p dA = \int y (\rho g h) dA \\ &= \int y (\rho g y \sin \theta) dA \\ &= \rho g \sin \theta \int_{y_A}^{y_B} y^2 dA \end{aligned}$$

But

$$\int_{y_A}^{y_B} y^2 dA = I_0 = \begin{array}{l} \text{moment of inertia of} \\ \text{surface } AB \text{ about the axis} \\ \text{at the watersurface} \end{array}$$

$$\begin{aligned} \therefore y_F F_R &= \rho g \sin \theta I_0 \\ &= \rho g \sin \theta (I_c + \bar{y}^2 A) \end{aligned}$$

where I_c = moment of inertia of surface AB about its centroid

A = area of surface AB .

Now $F_R = \rho g \sin \theta \bar{y}A$ from equation 2-50

$$y_F = \frac{\rho g \sin \theta (I_c + \bar{y}^2 A)}{\rho g \sin \theta \bar{y}A}$$

$$y_F = \frac{I_c}{\bar{y}A} + \bar{y}$$

or

$$y_F - \bar{y} = \frac{I_c}{\bar{y}A} \quad (2-51)$$

Thus, the resultant force on an inclined surface acts at a distance below the centroid of the surface equal to the moment of inertia of the surface about the centroid divided by $\bar{y}A$. Note that for a given surface area A , the distance the resultant force acts below the centroid diminishes with increasing depth.

Example 2-8: Consider the hydrostatic forces acting on the side of a ship hull as illustrated in figure 2-28. Compare the resultant force acting on a 6-ft by 6-ft section and its point of application for (a) the section located on the surface ($\bar{h}_1 = 3$ ft), and (b) the section located with the centroid 15 feet below the surface. The specific weight for sea water is $\rho g = 64$ lb/ft³.

Solution: For vertical surfaces, it is customary to replace y with the depth h as indicated in the sketch.

$$F_1 = \bar{p}_1 A = \rho g \bar{h}_1 A$$

$$= 64(3)36$$

$$= \boxed{6,912 \text{ lb}}$$

$$h_{F_1} = \bar{h}_1 + \frac{I_c}{\bar{h}_1 A}$$

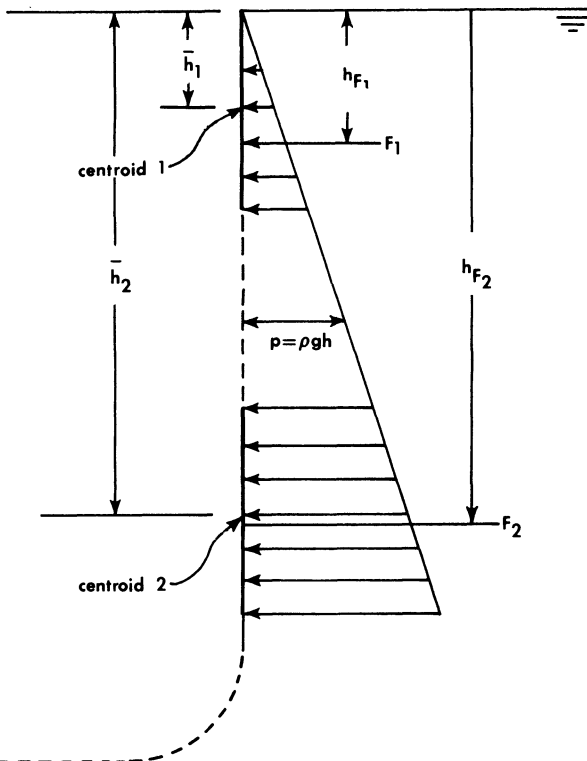


Figure 2-28. Forces on a ship hull

$$I_c = \frac{bh^3}{12} = \frac{6(6)^3}{12} = 108 \text{ ft}^4$$

$$h_{F_1} = 3 + \frac{108}{3(36)}$$

$$= \boxed{4 \text{ ft}}$$

which coincides with the depth of the centroid of the pressure-versus-depth triangle in figure 2-26.

$$F_2 = \bar{p}_2 A = \rho g \bar{h}_2 A$$

$$= 64(15)36$$

$$= \boxed{34,560 \text{ lb}}$$

$$h_{F_2} = \bar{h}_2 + \frac{I_c}{\bar{h}_2 A}$$

$$= 15 + \frac{108}{15(36)}$$

$$h_{F_2} = \boxed{15.2 \text{ ft}}$$

It can be shown that this depth also coincides with the centroid of the pressure-versus-depth trapezoid for the lower surface as illustrated in figure 2-26.

3. Forces on submerged curved surfaces.

Since the pressure forces always act normal to the surface it is possible to find the resultant forces by finding the horizontal and vertical components of the resultant force.

- The horizontal component of the hydrostatic force on a curved surface is equal in magnitude and in point of application to the resultant force on a vertical plane projection of the curved surface as illustrated in figure 2-29.
- The vertical component of the hydrostatic force on a curved surface is equal in magnitude to the weight of the liquid extending to the free surface vertically above the curved surface. It acts on a vertical line of action through the centroid of the volume of liquid above the surface. Although the horizontal component of the hydrostatic force can be easily computed, it is seen that the vertical components will require integration techniques, both to find the magnitude of the force and to locate the centroid of the volume.

ARCHIMEDES' PRINCIPLE

The techniques just developed can be used to demonstrate Archimedes' Principle, which can be stated as

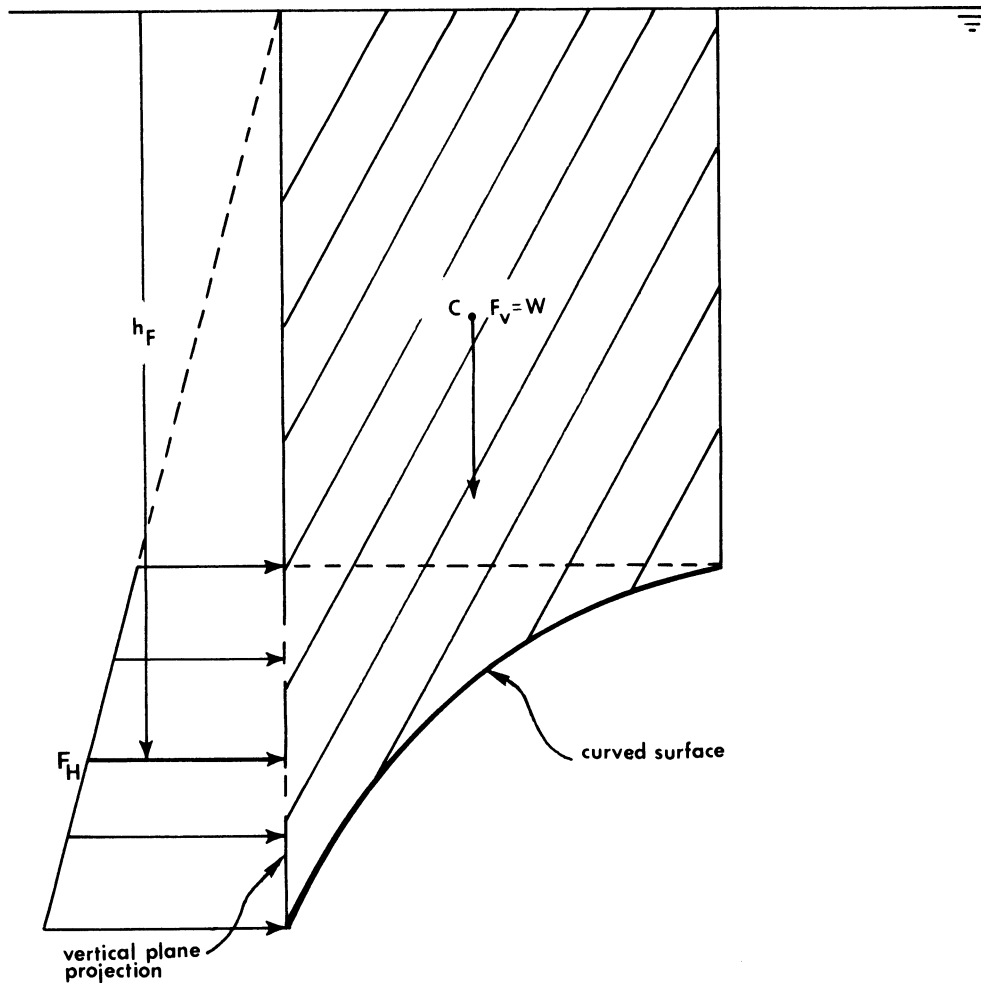


Figure 2-29. Forces on a curved surface

follows: *A body partially or completely immersed in a fluid is buoyed up, or sustained, by a force equal to the weight of fluid displaced.*

Consider the ellipsoidal body immersed in a liquid as illustrated in figure 2-30. It is seen from the figure that the horizontal pressure forces cancel each other since they both act on identical projected areas. The vertical force acting downward on surface ABC is equal to the weight of liquid directly above ABC or

$$F_{ABC} = \rho g (\text{volume } EABCF)$$

where ρ = density of the fluid.

The vertical force acting upward on surface ADC is equal to the weight of liquid directly above ADC or

$$F_{ADC} = \rho g (\text{volume } EADCF)$$

The net force or buoyant force acting on the surface $ABCD$ is thus

$$F_{ADC} - F_{ABC} = \rho g (\text{volume } EADCF - \text{volume } EABCF)$$

or

$$\begin{aligned} \text{buoyant force} &= \rho g (\text{volume } ABCD) \\ &= \text{weight of liquid displaced.} \end{aligned} \quad (2-52)$$

Volume $ABCD$ is called the displaced volume of the body, and equation 2-52 holds for both submerged and floating bodies. The buoyant force acts through the centroid of the displaced volume.

Problems

2-1. Determine the tension in the cable in figure 2-a. Consider the beam to be weightless.

2-2. Determine the tension in each of the cables in figure 2-b.

2-3. Determine the reactions at A and B for the

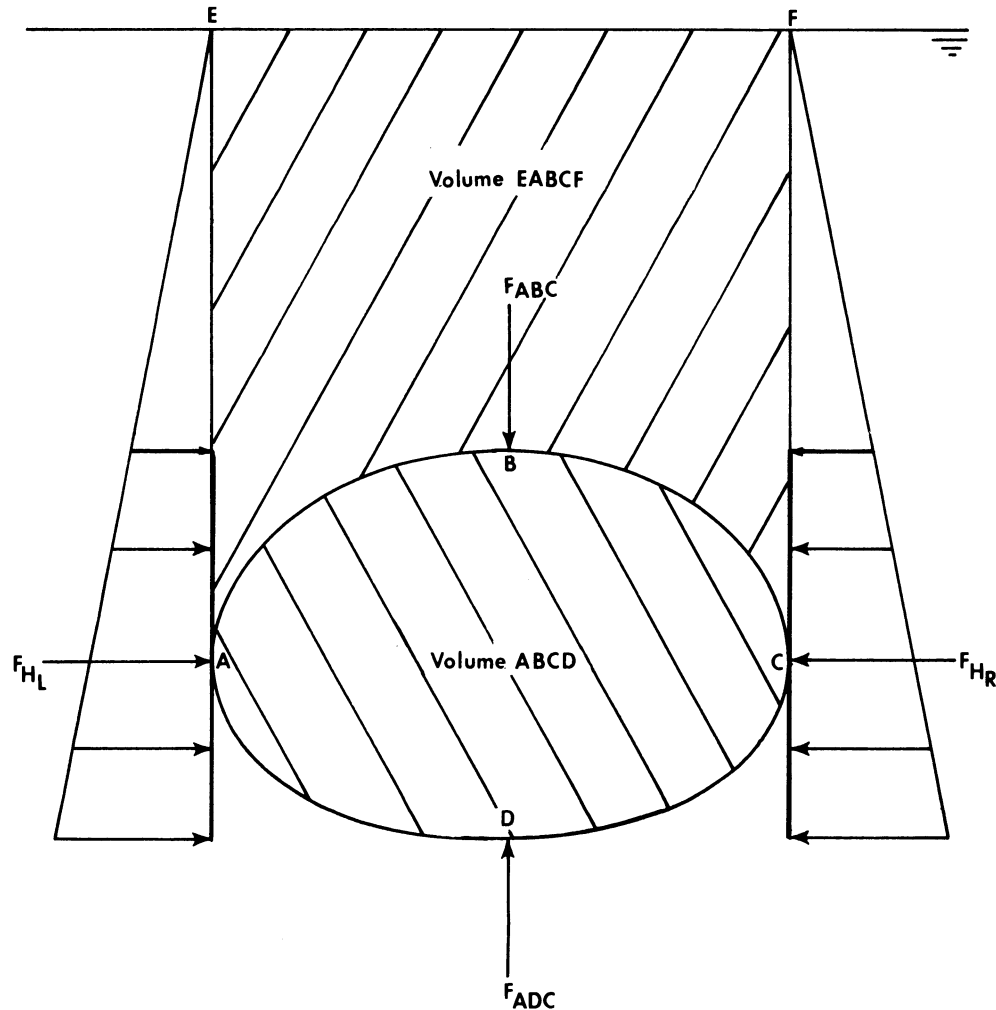


Figure 2-30. Forces on a submerged body

simply supported beam in figure 2-c. Consider the beam to be weightless.

2-4. Determine the reactions at point A for the cantilevered weightless beam shown in figure 2-d.

2-5. A ship is moored to a pier and can be boarded via a 30-ft-long brow weighing 500 pounds connected to the quarterdeck by a pinned joint at point A in figure 2-e. The other end of the brow rests on rollers on the pier at point B, ten feet below the quarterdeck. Two sailors are standing on the brow at the locations shown in the sketch. Determine the reactions at A and B.

2-6. Calculate the moment of inertia of a triangle about its centroid.

2-7. Calculate the moment of inertia of the V-block in example 2-4 about its base.

2-8. Calculate the centroid of the triangle in example 2-3 by using a horizontal strip for dA and integrating with respect to y .

2-9. Develop a general expression for the moment of inertia of an I-beam of width w , thickness t , and height h about the vertical and horizontal centroidal

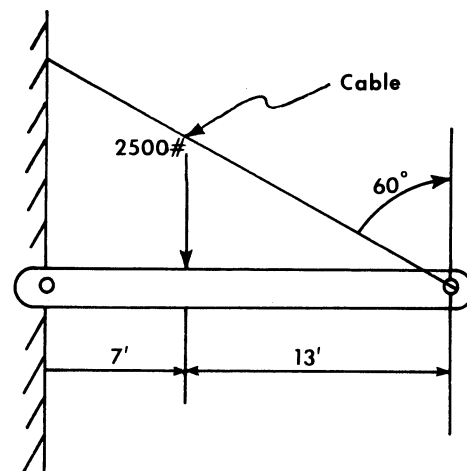


Figure 2-a.

axes. Compare these expressions with those for a channel section of the same dimensions.

2-10. Calculate the radius of gyration of the I-beam in example 2-7.

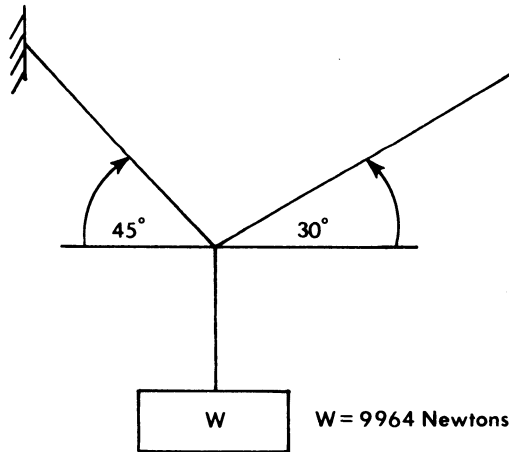


Figure 2-b

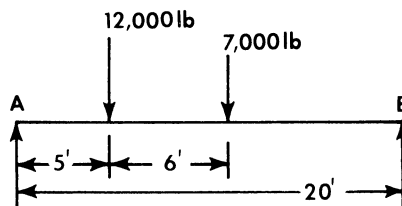


Figure 2-c

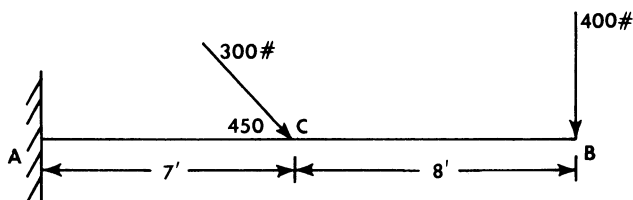


Figure 2-d.

2-11. If the pressure gauge on a steam condenser indicates a reading of 24 inches of mercury vacuum, what is the absolute pressure in psi if the barometer reads 750 mm of mercury?

2-12. Prove that the resultant force F_2 in example 2-8 acts at the centroid of the pressure-vs-depth trapezoid for the lower surface in figure 2-26.

2-13. A rectangular gate 6 feet high and 4 feet wide is hinged as shown in figure 2-f and is used to control the height of water in the reservoir. A spring is provided to hold the gate closed against the stops if the water is below a certain level. By adjusting the spring tension, the gate can be made to open at a given water depth. Find a) the horizontal force on the gate

b) the location of the resultant horizontal force on the gate

c) the spring force required to just keep the gate closed if the water is at elevation 9 feet but that will let the gate open for greater water elevations.

2-14. An irregularly shaped volume of a new ma-

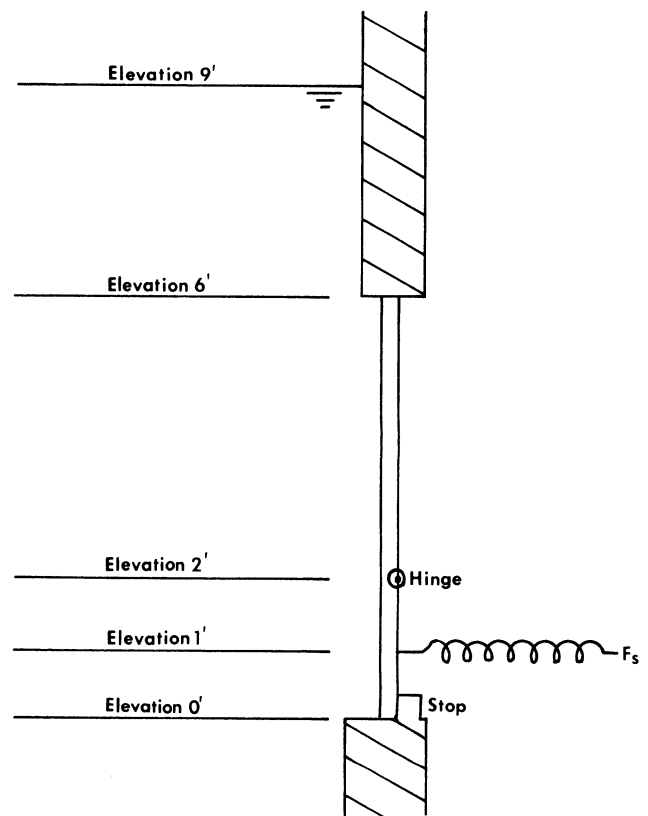


Figure 2-f

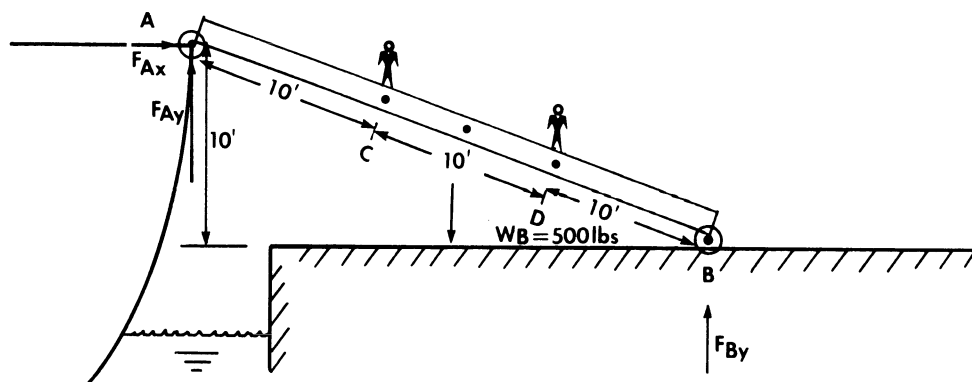


Figure 2-e

terial is attached to a scale and lowered into a tank of fresh water, as illustrated in figure 2-g. When the body is completely submerged, the scale reads 35 pounds and the level of the water in the 1.0-ft-diameter tank has risen 1.0 inch. Determine the specific gravity and the density of the material.

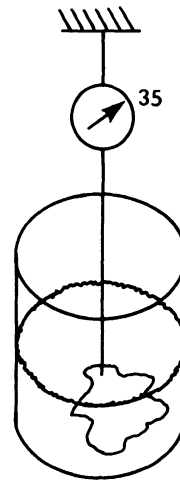


Figure 2-g.

Ship Geometry and Hydrostatics

3-1 Ship Geometry

Considering the complexities and critical designs of the many systems and subsystems within a ship, there is no feature as important to a ship as its geometric configuration.

The shape of a ship's hull determines the power required to drive it; it reflects directly the ship's speed; it determines the quantity of payload and the comfort and habitability within the ship; more important, it largely establishes the limits of safety and stability as well as the motion of the ship in a seaway. All of these factors, as was seen in the previous chapter, form the critical substance of the ship system. Before proceeding further with the methods and theory of ship design, some familiarity with ship geometry must be established.

LINES

To delineate a three-dimensional complex form such as a ship's hull and present it in a two-dimensional medium, such as a scaled drawing, one must resort to a direct application of descriptive geometry. In the case of the ship, the three basic projections become the *sheer plan*, the *half-breadth plan*, and the *body plan* (figure 3-1).

One can begin with the following analogous description. Suppose a ship's basic hull shape is placed in an imaginary rectangular box whose bottom and sides just touch the ship's surface. The bottom of this box may be used as a reference base and called simply the *base plane*. Now, if one also imagines planes parallel to this base plane slicing through the ship's hull form at regularly spaced vertical intervals, one has the familiarly known *waterlines*. It does not matter whether the ship will float at any of these levels; these intersections or traces are only used for references and delineate the horizontal segments of the hull from the bottom to the upper deck levels.

The waterlines are generally spaced every one or two feet and numbered from the keel or *base line*, the centered line of the base plane where it touches the ship. One of the specially designated waterlines is the *designed waterline (DWL)*, where the ship is designed to float at a predetermined load.

Now imagine planes that slice down through the hull parallel to the ends of the box and cut the hull like bread slices. These intersections are called simply *sections*. They are located at regular intervals; the locations are called *stations* and are numbered from forward aft. The first section is called the *forward perpendicular (FP)* and the aftermost one, the *after perpendicular (AP)*. The exact location of these perpendiculars is of some importance because the *length between perpendiculars (LBP or L_{PP})* is significant. In naval-design practice, the length between perpendiculars is customarily coincident with the DWL. For merchant-ship practice, the location of the after perpendicular most frequently is coincident with the vertical rudder post. In all cases, the forward perpendicular is coincident with the forward extremity of the DWL. This is also called the zero station. For ships with a bulbous bow, a distinction is made between the *length overall (L_{OA})* and the *length overall, submerged (L_{OS})*. The various characteristic ship lengths are illustrated in figure 3-2.

Normally there are ten station spaces numbered from zero through ten, making, in all, eleven basic sections. In larger ships and in naval combatants the number of station spaces is frequently doubled or at a greater multiple of ten. The section halfway between the first and last section is referred to as the *midship section* and designated by the symbol $\boxtimes$. In many large commercial and some naval ships, the actual shape of this section does not vary for some distance forward and aft of the midship section. This section, in such ships, is called the *parallel middle body*.

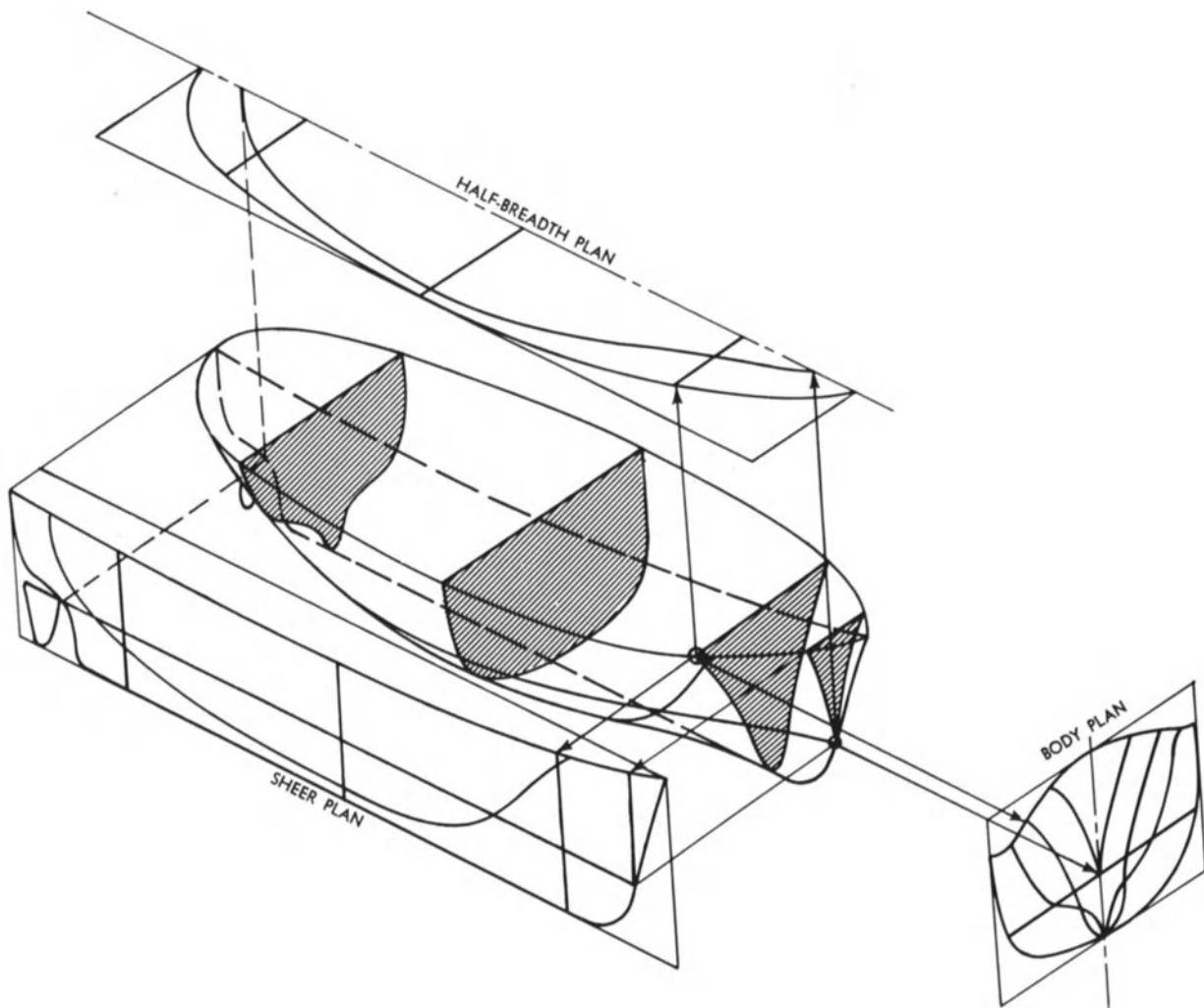


Figure 3-1. Projection of ship's lines

A third series of planes is referred to the central plane of symmetry of the ship. A longitudinal plane splitting the ship equally and vertically from stem to stern is the *centerline plane*. This plane must contain the vertical profile of the ship, including the external stem contour and stern and keel contours showing the rudder and appendage profiles as well as the main deck at center. Parallel to this centerline plane are regularly spaced reference planes cutting the ship's hull along lines called *bow and buttock lines* or *buttock lines* or simply *butts*. The spacing interval of the buttock lines is up to the designer, but usually it is a basic unit dimension of one foot, two feet, or four feet. It is useful to space the buttock lines so that one falls near a distance equal to one-half of the half-breadth of the ship from the center plane. This is referred to as the *quarter beam buttock* and is useful in relating the ship's curvature to similarly located reference lines on other ships.

Returning to the imaginary circumscribing box, one

can now visualize more simply three basic projections of the ship's form. The side of the box can be used as a screen on which to project the ship's profile, showing all of the profile shapes and buttock lines. If the ship's surface were transparent and the lines of intersection of the reference planes were opaque, with a projecting lamp like the sun at an infinite distance, the shadow outline on the screen would form the *sheer plan*. Likewise, the shadow form on the bottom of the box would be the *half-breadth plan* (being symmetrical, only one-half is needed, and so it is named).

On the end of the box, the projection is basically the *body plan*. However, to avoid confusing intersections of the many section shapes, the forward sections are shown to the right and the after sections to the left of the centerline. In the above manner, it is possible to describe a ship's systematized *lines drawing*, which is remarkably universal in its application. Even with our computerized technology, there is as yet no acceptable

substitute for, or alternate means of, delineating the huge surfaces of a ship's form in a more satisfactory way than the lines drawing.

After the ship has been designed and its hull form determined and graphically described as above, it is customary to set up a matrix system that a digital computer can cope with. This matrix, arranged in tabular form and called a *table of offsets*, is taken from the scaled lines drawing, expressed in full-size dimensions, and faired by computer. *Fairing* is nothing more than the refining and smoothing of the minor lumps, inconsistencies, and inaccuracies in the drawing. Mathematically, a *fair line* is one which has a continuous second derivative.

The fairing process is necessary, particularly in large vessels where the original design work may have been done at a scale whose ratio to full size was 1:100. Small discrepancies, such as the width of a pencil line in the design process, take on significant magnitude when blown up to full size.

The lines drawings for a typical World War II destroyer (DD 692) are shown in figure 3-3. Figure 3-4 demonstrates a method for emphasizing the body plan and fore-and-aft profiles in technical reports that lack the page size to show the full lines drawings.

THE MEASURE OF A SHIP

Before proceeding further in the technology of ship design, some of the language and terminology applying both to geometry and flotation properties must be resolved.

The first of these definitions most properly concerns size or three-dimensional magnitude. The popular layman's description of a ship's size is simply a statement of its length. Such a description is not only grossly inadequate, it is inaccurate. A linear dimension such as length is one dimensional. An eight-oared rowing shell is actually longer than the average harbor tug boat, yet no one would dispute which vessel is larger.

There are several notations for the size of ships, and

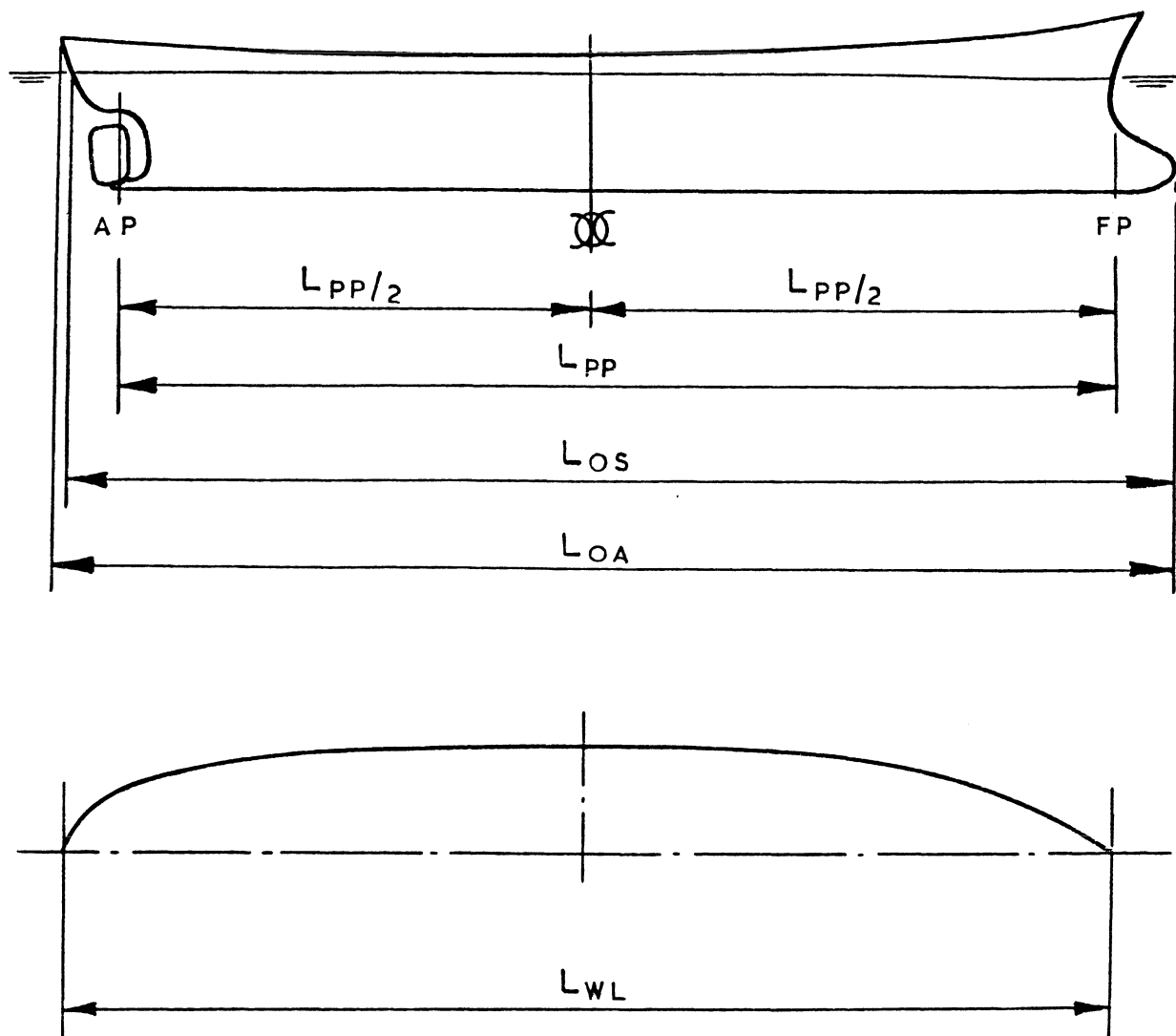


Figure 3-2. Characteristic ship lengths

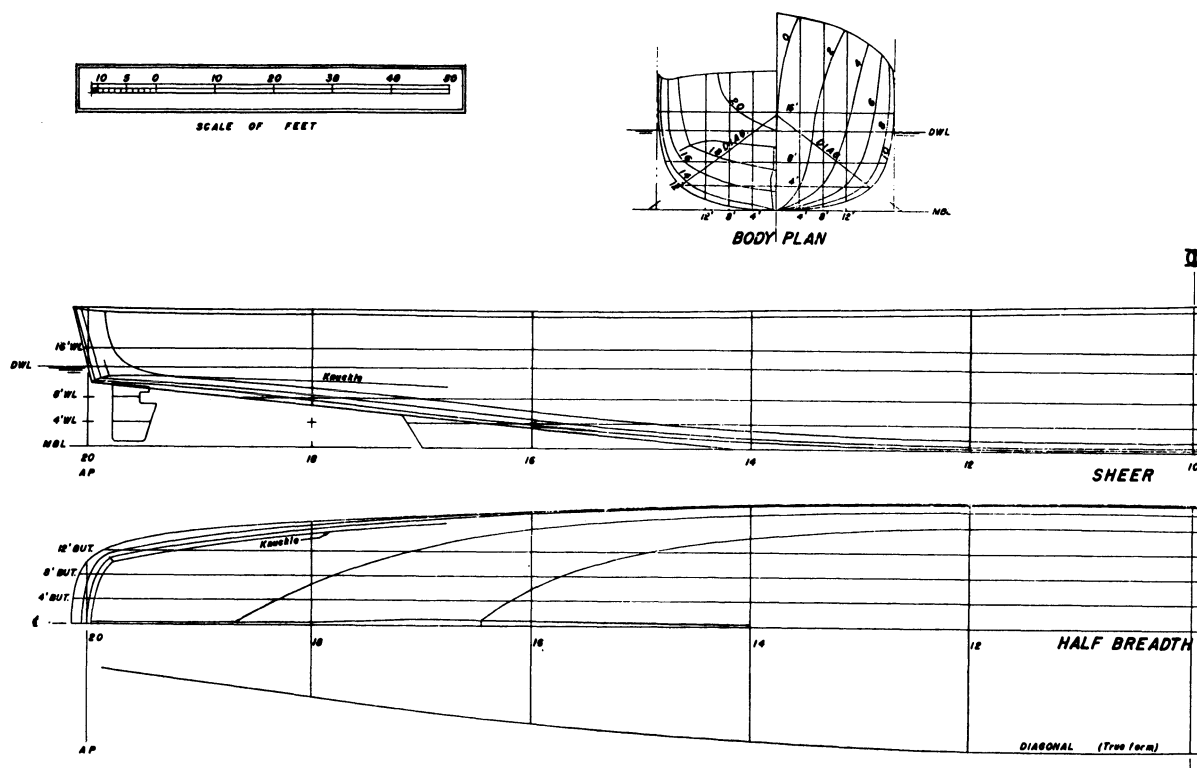


Figure 3-3. Lines drawing for DD 692

some can be confusing. Nevertheless, these denominations concern themselves with three dimensions. The two categories of size are weight and volume, which in our normal understanding may seem of opposite meaning. *Displacement* of a ship is a statement of its weight; *tonnage* is generally a measure of its volume or capacity.

Displacement. The weight of the water (salt water at 64 pounds per cubic foot) that the ship displaces when floating freely is called *displacement*. The waterline or drafts must be designated to indicate its loading condition, and therefore the weight of the ship is established equivalently for a given loading or condition of flotation, as shown in discussions of Archimedes' Principle elsewhere in this book. The symbol Δ is used for displacement and its magnitude is usually specified in English long tons.

Tonnage. The volumetric capacity of a ship, either gross or net, is described in cubic feet (divided by 100) and known as *registered tonnage*. Historically, a very important and standard cargo for European sailing vessels was wine, stored and shipped in casks called *tuns*. These tuns of wine, because of their uniform size and their universal demand, became a standard by which a ship's capacity could be measured. A tun of wine weighed approximately 2,240 pounds and occupied nearly 60 cubic feet. Port dues and fees were levied on the basis of this standard, but it quickly became evident that a

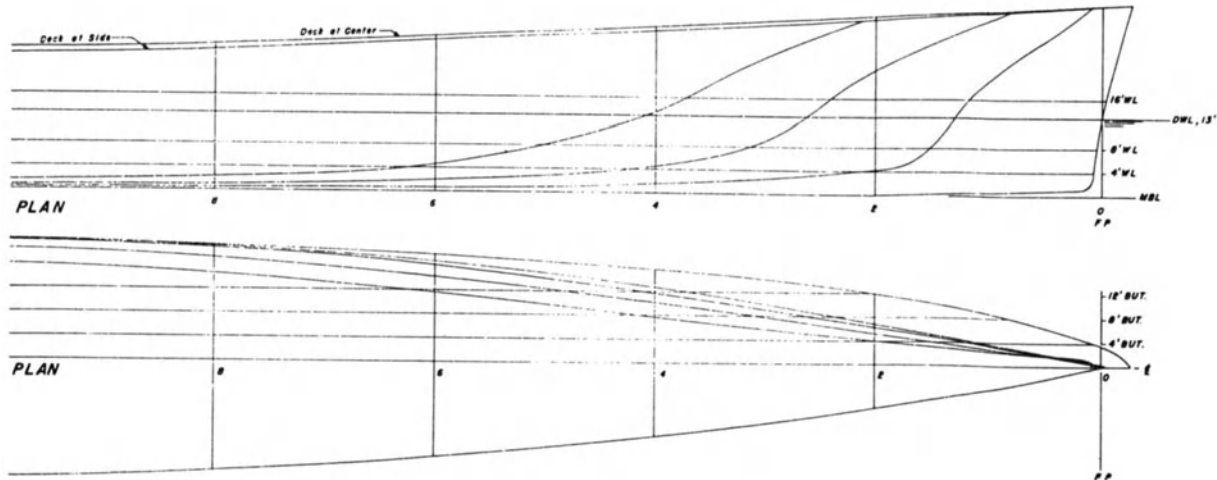
more equitable basis of measurement was needed to compare vessels of different construction and styles. The first standard measurement rules, established in England in the seventeenth century, consisted simply of the length of keel multiplied by the inside breadth and depth of the hold divided by a factor of 94. This formula, which became law, was considered a fair and uniform average indicating the number of tuns of wine that vessels of the period could carry. At this time, also, the word acquired its present spelling, "ton," and the standard weight of the so-called English long ton became 2,240 pounds. The long ton is the same today and is the ship designer's standard of weight. The long ton equals 1.01605 metric tons, the other international measure of cargo capacity. Since the two measures are so close in value, they are used somewhat interchangeably when discussing approximate weight-carrying capacities.

The subject of registered tonnage and measurement rules, which to a large extent are responsible for ship's harbor dues, fees, canal revenues, etc., is complex. Further knowledge in this area should be sought in such sources as *Ship Design and Construction* (Taggart 1980, chapter 5) or in various government shipping regulations.

DEADWEIGHT TONNAGE

There is one usage of the term *tonnage* to be noted before proceeding to other terms. The increasing size

PRINCIPAL DIMENSIONS	
LENGTH BETWEEN PERPENDICULARS	383' 0"
LENGTH OVERALL (MOLDED)	390' 6"
BREADTH, EXTREME (MOLDED)	40' 10 $\frac{1}{4}$ "
DEPTH AT CENTER (MOLDED)	23' 10"
DRAFT, DESIGNED WATERLINE (MOLDED)	13' 0"
DISPLACEMENT (MOLDED, SALT WATER)	3010 TONS



of large-bulk cargo carriers has, since World War II, made the term *deadweight tonnage* common. In this instance, tonnage does refer to weight and indicates the difference in displacement between the loaded and unloaded condition. Deadweight tonnage can be simply described as the cargo-carrying ability of a ship expressed as total weight. With this in mind, the term becomes most graphic when applied to the great tankers being built and conceived in a magnitude of several *hundred thousand* deadweight tons. The deadweight tonnage of ships of the World War II era seldom exceeded 20,000 tons.

DIMENSIONS

In the category of dimensions, it is necessary to define some terms used frequently in ship design. The following terms are therefore categorized, and their definitions considered precise and unbending:

1. *Molded hull form*—The surface of the ship's hull inside the planking, plating, or skin fabric. It is considered to be a smooth, faired form not subject to structural irregularities caused by plate laps, butt straps, appendages, etc. (figure 3-5).

2. *Drafts*—The *forward draft* and *after draft* are those vertical distances from the base line to the waterline of reference measured at the forward and after perpendiculars respectively. The *molded draft* is measured from

the molded base line. The *keel drafts* are measured from the bottom of the keel. *Draft scales* for keel drafts are placed on both sides of the ship as close as possible to the perpendiculars referenced. The *mean draft* is the arithmetical mean of the drafts forward and aft. The *designed draft* is the height of the designed waterline above the base line.

3. *Freeboard*—The vertical distance from the water or designated waterline to the weather-deck edge at any given location along the ship. The weather deck is the uppermost continuous deck with no overhead protection.

4. *Sheer*—The difference between the design trim freeboard at any point and that of the midship section. The *sheer line* is the line of intersection of the main or weather deck with the side of the ship.

5. *Camber*—This represents the curvature in an athwartship or transverse vertical plane; it is the height of the deck at the centerline above the height at the side.

6. *Tumble home*—The slant inward from the vertical of a transverse section of a hull above the designed waterline. It is the opposite of flare.

7. *Flare*—The slant upward and outward from the vertical of a transverse section of a hull above the designed waterline. Flare is the opposite of tumble home.

8. *Deadrise*—Otherwise known as rise of floor. The athwartship rise of the bottom from the keel to the bilge.

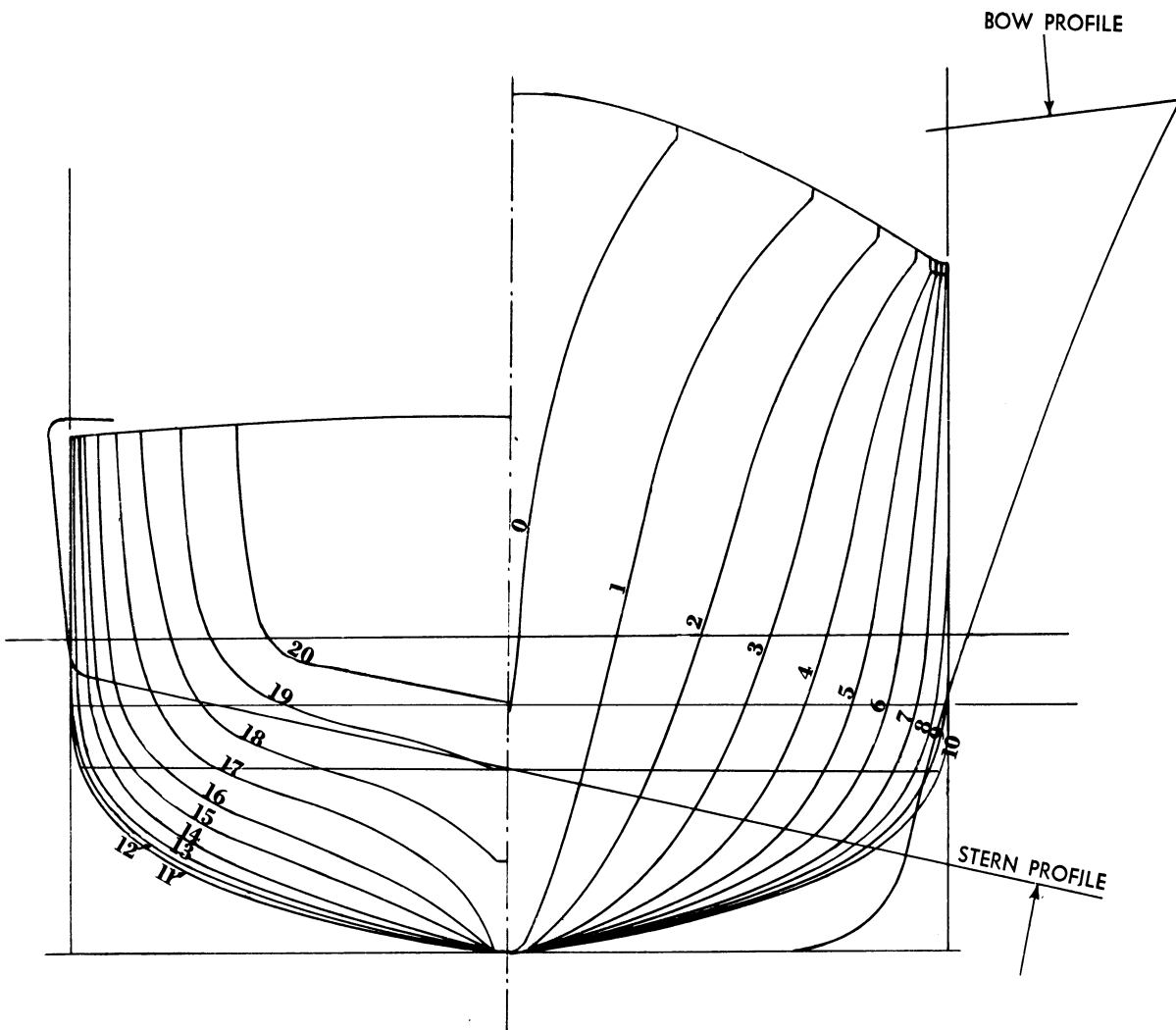


Figure 3-4. Typical body plan and fore-aft profiles (destroyer type). This simplified graphical expression of hull shape is frequently used as a substitute for the lines drawings in technical reports relating to ships.

FLOTATION CHARACTERISTICS

Trim. When a ship is not floating at the designed waterline or at a waterline parallel to it, then it is *out of trim*. The amount of trim is the difference between the drafts forward and aft. There are some ships for which the above definitions of trim must be modified, because they are designed with a keel that is not horizontal but sloped down aft. Such ships are said to have a designed *drag*; the amount of drag is the greater designed draft aft minus the designed draft forward. So the trim of a ship with drag is the difference between the drafts forward and aft in excess of the drag. In such cases, it is practical to assign positive and negative values to trim and indicate trim (down) by the stern or trim (down) by the bow respectively.

List, Heel, and Roll. Ships are, of course, designed to, and normally do, float upright—that is, with their longitudinal vertical center planes perpendicular to the water's surface. However, because of unsymmetrical

loading conditions or other unbalanced forces, they may incline transversely with respect to their normal upright positions. Such transverse inclinations are described as *list*, *heel*, or *roll*, depending upon the nature of the situation. *List* describes a definite attitude of transverse inclination of a static nature. *Heel* describes a temporary or semipermanent inclination, generally involving motion, whereas *roll* involves recurrent inclination from side to side.

A ship may *roll* about her angle of *list* in a seaway, *heel* away from her angle of *list* while in a turn, or *heel* for more or less enduring intervals due to strong wind pressure.

3-2 Form Coefficients

In order to refer to certain proportions of ships—to compare them in form with no regard to their actual dimensions or difference in dimensions, to describe their shapes more precisely than as “fat” or “thin,” “full” or “fine”—certain geometric qualities can be related as

ratios or dimensionless coefficients. These coefficients of form are very useful in comparing certain performance characteristics associated with hydrodynamic phenomena.

In the following relationships, the symbols used are defined as follows:

- L —Length on the designed waterline, length between perpendiculars, or length overall, submerged as appropriate to the computation
- T —Draft to the designed waterline, or draft at the maximum section area, as appropriate to the computation
- B —Beam amidships at the designed waterline, or beam at the maximum section area, as appropriate to the computation
- ∇ —Volume of displacement at draft T
- A_M —Area of the midship section at draft T
- A_X —Area of the maximum section to the designed waterline
- A_w —Area of the waterplane at draft T
- Δ —Displacement tonnage at draft T

(Please note that the above dimensions may be either molded dimensions or nonmolded, as appropriate to the hull and structural materials.)

The coefficients most commonly used by naval architects are as follows:

1. *Block coefficient* (C_B)—The ratio of the volume of displacement to the volume of rectangular block having a length appropriate to the type of ship and a beam and draft equal to that at the maximum section area.

$$C_B = \frac{\nabla}{LBT} \quad (3-1)$$

The block coefficient relationships are illustrated in figure 3-6.

2. *Prismatic coefficient* (C_P)—The ratio of the volume of displacement to the volume of a cylinder having a length L and a cross section equal in area to that of the maximum section at the designed waterline.

$$C_P = \frac{\nabla}{LA_X} \quad (3-2)$$

The prismatic coefficient relationships are illustrated in figure 3-7.

3. *Vertical prismatic coefficient* (C_{VP})—The ratio of the volume of displacement to the volume of a vertical cylinder having a draft T and a horizontal cross section equal in area to that of the waterplane area at draft T .

$$C_{VP} = \frac{\nabla}{TA_w} \quad (3-3)$$

Both longitudinal and vertical prismatic coefficients can also be referred to the different parts of the ship, such

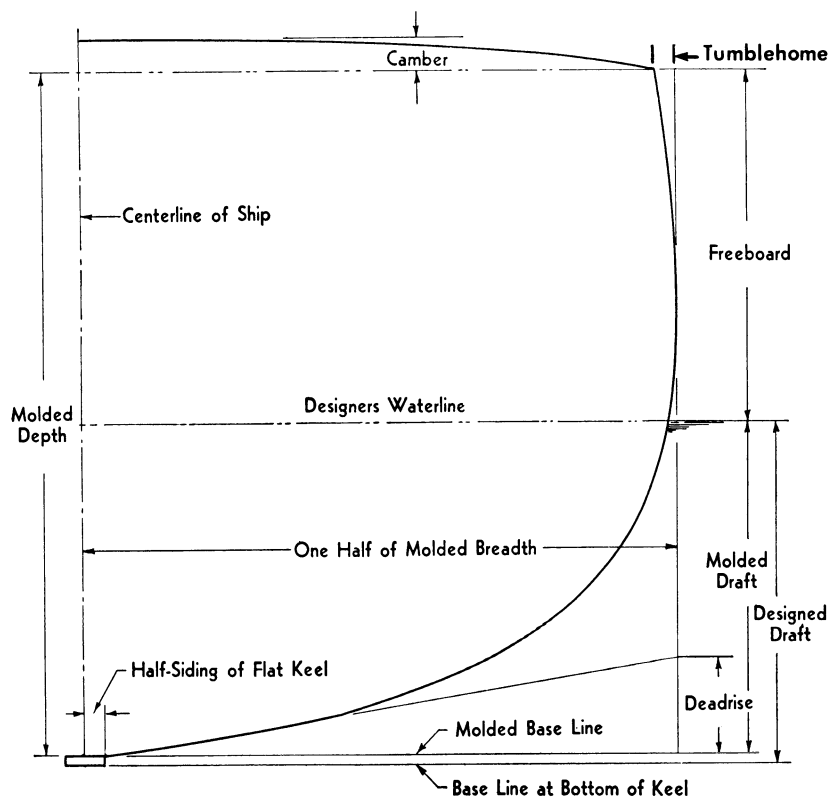


Figure 3-5. Midship-section molded-form definitions

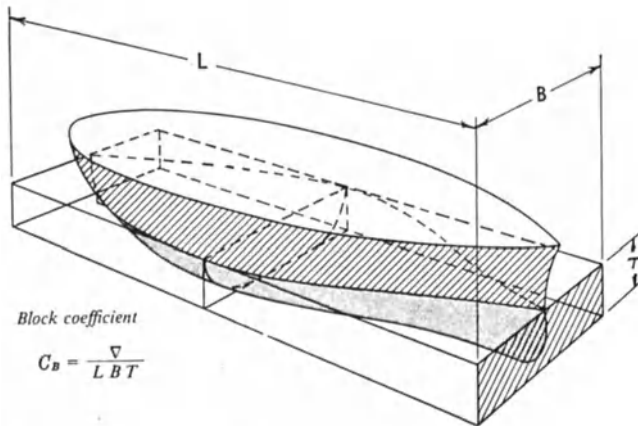


Figure 3-6. Block coefficient relationship

as the afterbody, forebody, entrance, and run. In any case, the assumed length, draft, and cross-section area, if different from the above, should be clearly indicated.

4. *Waterplane area coefficient (C_{WP})*—The ratio of the waterplane area at the designed or loaded waterline to the area of the circumscribing rectangle.

$$C_{WP} = \frac{A_W}{L_{WL} B} \quad (3-4)$$

The waterplane area coefficient is illustrated in figure 3-8.

5. *Waterplane inertia coefficients (C_{IT} and C_{IL})*—The ratio of the moment of inertia of the waterplane area to that of the circumscribing rectangle in the transverse and longitudinal directions.

$$\text{transverse } C_{IT} = 12 I_T / B^3 L \quad (3-5a)$$

$$\text{longitudinal } C_{IL} = 12 I_L / B L^3 \quad (3-5b)$$

where I_T = transverse second moment of area (or moment of inertia) of the waterplane

I_L = longitudinal second moment of area (or moment of inertia) of the waterplane.

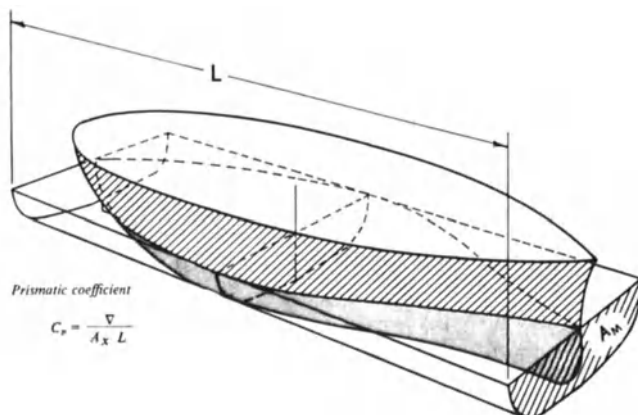


Figure 3-7. Prismatic coefficient relationships

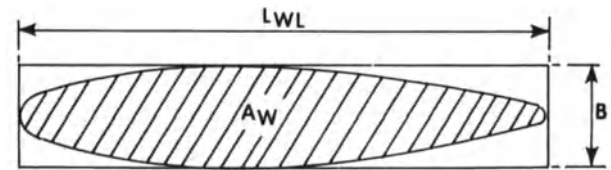


Figure 3-8. Waterplane coefficient

6. *Maximum transverse section coefficient (C_X)*—The ratio of the maximum transverse section area to the product of the beam and draft at this section.

$$C_X = \frac{A_X}{B_X T_X} \quad (3-6)$$

7. *Midship section coefficient (C_M)*—The ratio of the midship-section area to the area of a rectangle with width B and draft T .

$$C_M = \frac{A_M}{B T} \quad (3-7a)$$

The midship section coefficient is illustrated in figure 3-9. Note that for cases where the midship section represents the maximum section area of a ship (i.e., A_M equals A_X), the following relationship exists between the block coefficient, prismatic coefficient, and midship section coefficient:

$$C_B = \frac{\nabla}{L B T} = \frac{\nabla C_M}{L A_M} = C_P \cdot C_M \quad (3-7b)$$

There are also certain commonly used ratios of dimensions, and these with their approximate range of values for monohull displacement ships are

<i>length – beam ratio</i> L_{pp}/B	range, 3 to 12
<i>length – draft ratio</i> L_{pp}/T	range, 7 to 30 (except on specific types of hulls, such as deep fin or keel sailboats)
<i>beam – draft ratio</i> B/T	range, 1.8 to 4
<i>volumetric coefficient</i> ∇ / L_{pp}^3	(L is cubed in the expression to maintain a dimensionless relationship)
<i>displacement – length ratio</i> $\frac{\Delta}{(L_{pp}/100)^3}$	range, 50 to 500 (This ratio was devised by D. W. Taylor and differs from the preceding ratio only in its dimension. The number 100 is present merely to keep the values to convenient numbers)

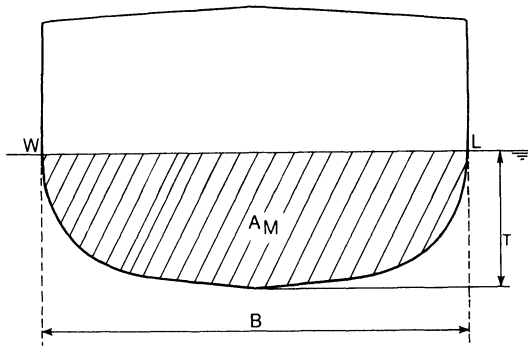


Figure 3-9. Midship-section coefficient

Some typical form coefficients of specific vessel types are given in table 3-1 for comparison.

LOCI AND CENTERS OF THE SHIP

Center of Buoyancy (B). The center of buoyancy is the line of action of the resultant of all the buoyant forces on the immersed portion of the ship's hull. It passes through the geometric center (centroid) of the underwater form, at which point it is called the center of buoyancy. The force of buoyancy acts vertically at this point, and for a ship floating at rest, the weight of the ship acts downward vertically through or in the vertical line containing the center of buoyancy. Its vertical position is designated *VCB* and longitudinal position *LCB*. The height of the center of buoyancy above the keel is designated $\overline{KB}$ and is determined by

$$\overline{KB} = \frac{1}{\nabla} \int A_w z dz$$

where A_w = area of a waterplane at height z above the keel.

Approximation for the Vertical Position of B

Because the various rules for approximation of integration (appendix B) do not lend themselves conveniently to ship forms in vertical layer integration, an

empirical method has been developed for locating the centers of volume in typical ships' forms. S. W. F. Moorish developed such an equation in 1892 that remains useful in most modern ship forms and provides a very close approximation for the vertical position of the center of buoyancy of a ship throughout a reasonable change of drafts. This formula is generally acceptable to designers for $C_x \leq 0.9$, but the position of *B* should be checked by a method using one of the rules of integration for at least two waterlines before plotting the values on the hydrostatic curves.

The formula, as first presented by Moorish, is

$$\text{distance of } B \text{ below waterline} = \frac{1}{3} \left(\frac{T}{2} + \frac{\nabla}{A_w} \right) \quad (3-8)$$

where T = molded draft (mean)

∇ = volume of displacement at T

A_w = area of corresponding waterplane.

The above equation may be modified to give the height of the center of buoyancy above the base line and is more convenient in this form. Thus,

$$\overline{KB} = \frac{1}{3} \left(\frac{5T}{2} - \frac{\nabla}{A_w} \right) \quad (3-9a)$$

$$\overline{KB} = \frac{1}{3} \left(\frac{5T}{2} - \frac{C_B L B T}{C_{WP} L B} \right)$$

$$\overline{KB} = \frac{T}{3} \left(\frac{5}{2} - \frac{C_B}{C_{WP}} \right) \quad (3-9b)$$

Another approximate formula of similar scope, but in some cases giving values closer to the more precisely calculated values by computer-based integration, is

$$\overline{KB} = T \left(\frac{A_w}{A_w + \nabla/T} \right) \quad (3-10a)$$

$$\begin{aligned} \overline{KB} &= T \left(\frac{C_{WP} L B}{C_{WP} L B + \frac{C_B L B T}{T}} \right) \\ &= T \left(\frac{C_{WP}}{C_{WP} + C_B} \right) \end{aligned} \quad (3-10b)$$

This equation is quite useful for $C_x \geq 0.9$.

Center of Gravity (G). The center of gravity is the center through which all the weights constituting the ship and its contents may be assumed to act. This center has the conventional meaning used in mechanics when it applies to the whole ship; i.e., it is the point at which the sum of the moments of all the weights in the ship with reference to any axis through this point is equal to zero.

Table 3-1. Typical Form Coefficients

Coefficients	Destroyer	Cargo Liner	Harbor Tug	Great Lakes Bulk Freighter	Passenger Liner
C_B	0.52	0.64	0.58	0.87	0.59
C_M	0.83	0.96	0.89	0.99	0.95
C_P	0.62	0.66	0.65	0.88	0.62
C_{WP}	0.74	0.76	0.80	0.91	0.72
Ratios					
L/B	9.82	6.92	4.18	9.67	8.38
L/T	32.75	16.82	9.33	29.00	26.25
B/T	3.33	2.43	2.23	3.00	3.14

On ships of usual form, the center of gravity is near the waterline and the midship section. The weight of the ship may be considered to be concentrated here with the total gravitational force acting downward through it. For a ship floating at rest, the center of gravity lies in the same vertical line as the center of buoyancy. This is an important consideration in design calculations for trim and equilibrium. The vertical position of the center of gravity is with reference to the bottom of the keel amidships. It is designated $\overline{KG}$, and its longitudinal position LCG .

Center of Flotation (F). The center of flotation is the geometric center (centroid) of the waterplane area at which the ship is floating. It should not be confused with the center of buoyancy, with which it has no direct relation. It is the location of the axis about which the ship trims. Its position, measured as the distance from midships or from the fore or after perpendicular, is called the *longitudinal center of flotation* (LCF).

The Metacenter (M). When a ship rolls, heels, or is inclined from its position of equilibrium, the center of buoyancy moves to the new location of the centroid of the immersed volume as inclined. When the ship is inclined transversely, the center of buoyancy moves out from the centerline in a pseudo-elliptical path. The metacenter is defined as the intersection of the vertical through the center of buoyancy of an inclined body or ship with the upright vertical when the angle of inclination approaches zero as a limit. This is illustrated in figure 3-10. The metacenter is used as a convenient reference point and indicates, when referred to the cen-

ter of gravity, the degree of initial stability (see chapter 6).

For larger inclinations, the locus of the center of curvature of the instantaneous locations of the center of buoyancy is called the *metacentric* or *pro-metacenter* (see figure 3-11).

The *transverse metacenter* (M) conforms to the above definition for transverse inclinations. When the inclination is longitudinal (i.e., change of trim), the reference point generated as above is called the *longitudinal metacenter* (M_L). Its value is primarily that of setting up the equations for changes in trim and for intact and damaged stability analysis. In subsequent discussions, the use of the term metacenter will assume the transverse metacenter. The longitudinal metacenter will be specifically stated where it is referred to. The distance from the keel to the metacenter is designated by $\overline{KM}$ or $\overline{KM}_L$.

Metacentric Height ($\overline{GM}$). The metacentric height is the vertical distance measured on the ship's vertical centerline between the metacenter and the center of gravity.

Metacentric Radius ($\overline{BM}$). The metacentric radius is the distance between the center of buoyancy B and the metacenter M .

RELATIVE LOCATION OF CENTERS

The relative locations of the centers listed above are illustrated in figure 3-12. From this figure it can be seen

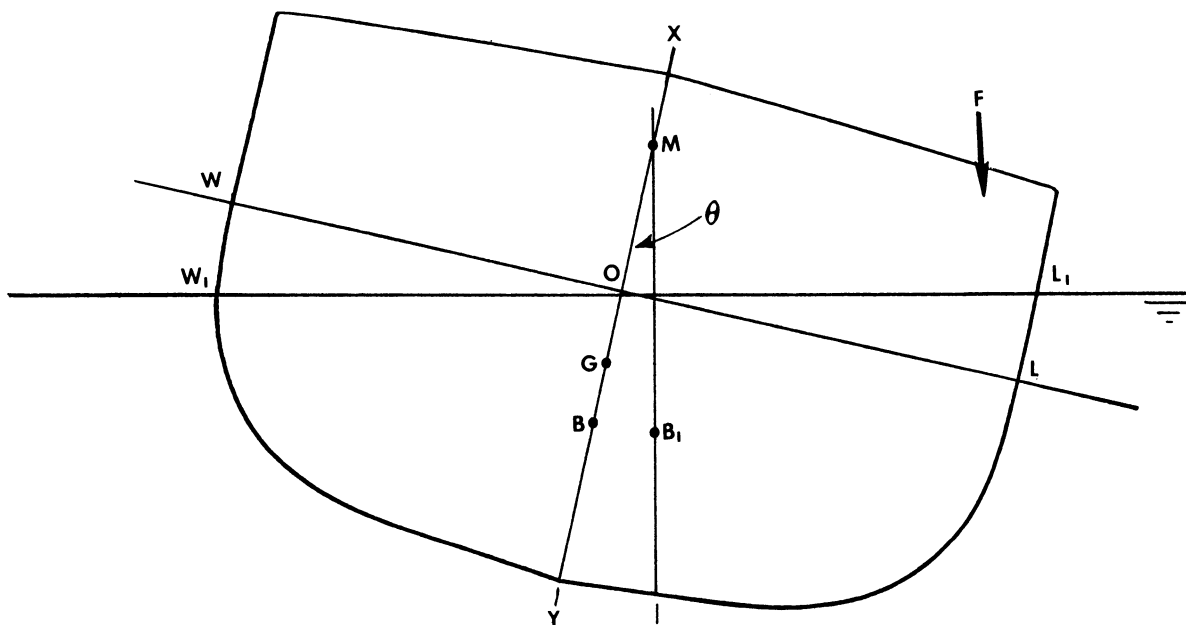


Figure 3-10. The metacenter M

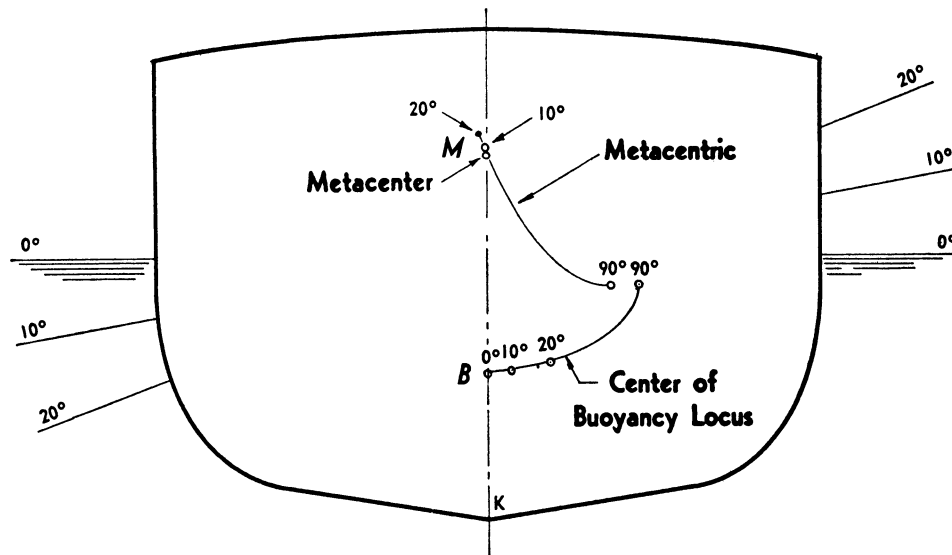


Figure 3-11. The metacentric

that the following relationships hold:

$$\overline{KM} = \overline{KB} + \overline{BM} \quad (3-11)$$

$$\overline{GM} = \overline{KM} - \overline{KG} \quad (3-12)$$

Longitudinal Characteristics. The relative locations of the longitudinal centers can likewise be stated as follows:

$$\overline{KM}_L = \overline{KB} + \overline{BM}_L \quad (3-13)$$

$$\overline{GM}_L = \overline{KM}_L - \overline{KG} \quad (3-14)$$

3-3 Ship Forms

The potential variety of ship-hull forms is as infinite as the possible variety of ships, the imaginations of their designers, and the requirements of their owners. The following discussion will be restricted to the more common hull forms.

The most basic description of a common ship or boat form is that of an opened envelope (figure 3-13). Such a description is inadequate, but it serves as a reference. If this opened envelope, with its vertical ends (the stem and stern) and bulging sides, was slit open at the bottom, it would have a foil shape with regular sections. Let us assume a symmetrical foil shape as a second state of development, as in figure 3-14. With a top and bottom on the foil (deck and hull bottom), it would be evident when the form is pushed through water that the edges between the sides and bottom should be rounded or cut away to allow an easier flow of water past and under the bottom. This, then, becomes the basic ship form, and the extent of rounding and fairing of the vertical surfaces into the flatness of the bottom becomes a matter of degree and compromise (figure 3-15).

Further modifications of this basic form are found desirable and almost always more expensive. The most economically built ships in the bulk cargo service, with

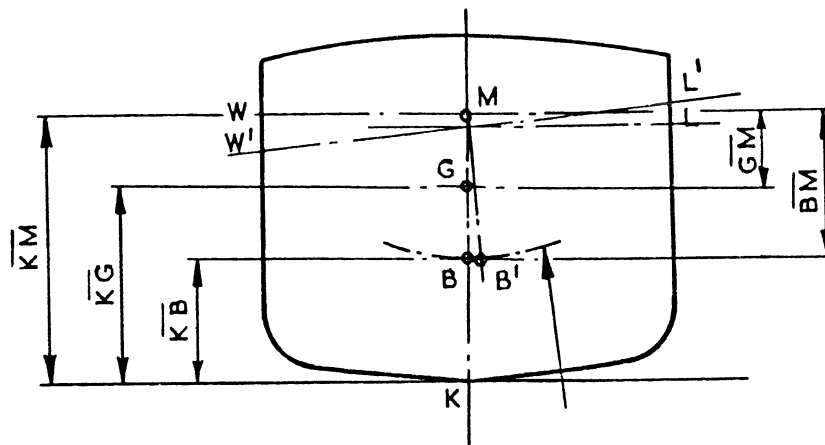


Figure 3-12. Transverse metacentric parameters

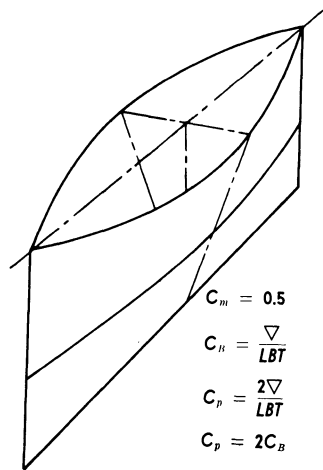


Figure 3-13. Simple envelope hull

their low power and speed, are similar in character to the simple shape described above.

For greater seaworthiness, better speeds, drier decks, and better motion in a sea, various changes are made in the basic form. The stem is angled or raked forward with a corresponding extension of deck space. The deck area also becomes widened when the bow sections are flared out with the resultant concavity typical of flared bows. Farther aft, the flatness of the bottom is given a gentle rise and transverse slope bending into a broader stern (figures 3-16, 3-17).

The extent of the fairing, rounding, and hollowing frequently produces ship forms that seem to have very little similarity to the basic shape described above. This is so apparent that some designers refer to another geometrical reference shape. This is the *double wedge*, where the vertical, sharp edge of the wedge is forward (the bow), and the horizontal or 90° turned wedge is aft and truncated, becoming the flat stern found in many vessels (figure 3-18). This basic shape is perhaps more adaptable to smaller higher-speed vessels. However, the double wedge is also recognizable in many large

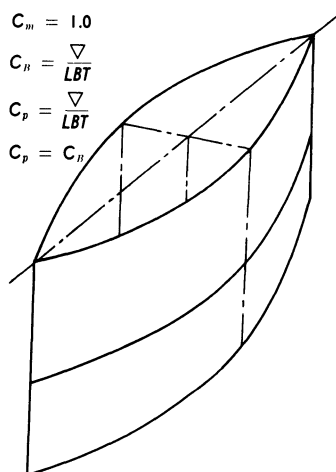


Figure 3-14. Foil hull

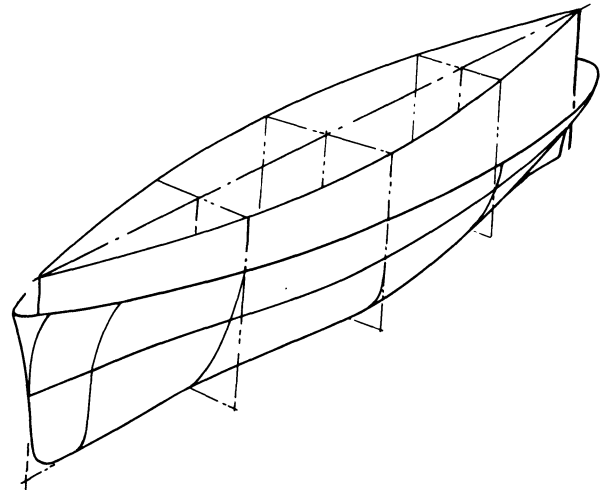


Figure 3-15. Emerging hull form

warships and in more modern auxiliary and naval support vessels (figure 3-19).

There are two portions of the ship's below-water hull form that concern the designer in his search for speed and economy through efficient hydrodynamics. These are the *entrance* and the *run*. The entrance is that portion of the hull from the leading edge of the stem aft to the section of greatest breadth (not necessarily midships). This entrance should very gradually widen with little or no noticeable change in curvature. Hollow waterlines that are sometimes evident in this portion of the ship often become disadvantageous because they create "shoulders" that cause secondary wave-making disturbances.

In slow-speed, full-bowed ships with high block coefficients (C_B), the entrance becomes even more difficult

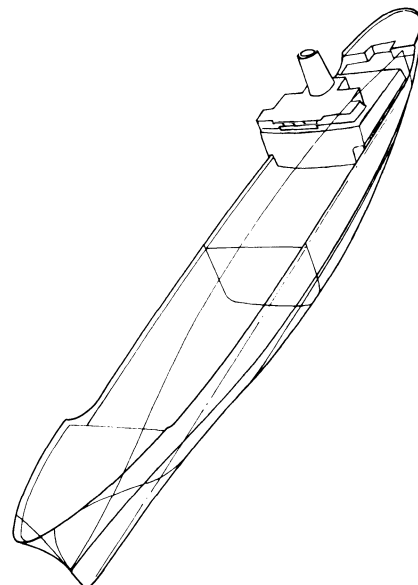


Figure 3-16. Ship-hull form: basic modern cargo liner



Figure 3-17. The USS *Severn* (AO 61), a typical hard-working fleet oiler. This is a ubiquitous form of hull found in most of the bulk carriers and cargo vessels of the world. Vessels of this type have extensive parallel middle bodies.

to design. A most promising and evident solution to this problem on such ships in recent years is a forward extension of the stem below the water. This form, evolving from the ancient ram bow to the “Taylor Bulbous Bow” and to its present elongated form, the “Inui Bow,”

has become a common characteristic in supertankers, other cargo vessels, and warships.

The second concern in below-water hull design is the *run*. The run is that portion of the bottom aft from the widest section to the trailing edges of the stern. Like

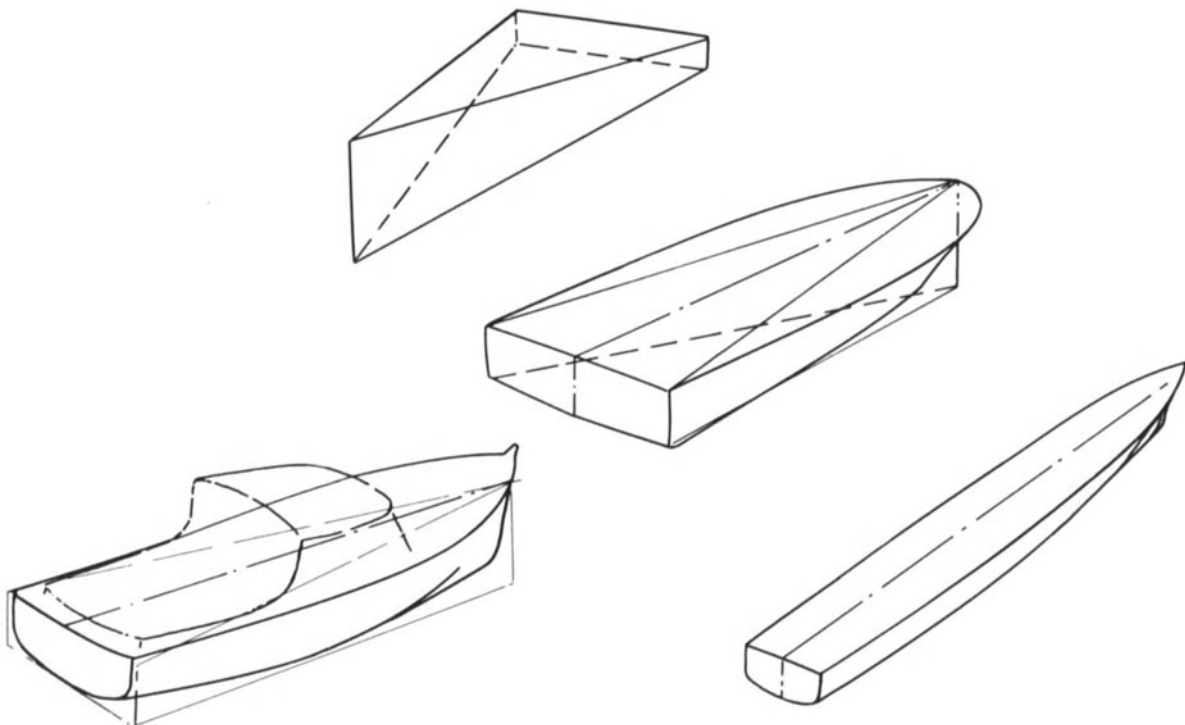


Figure 3-18. The double-wedge hull



Figure 3-19. The Soviet helicopter ship *Moskva*, a further example of the application of the double wedge to the requirement of both deck space and speed

the entrance, the run should have no quick changes of curvature. The curvature, where it exists, should be gradual and, in the final quarter of the hull, should be nearly flat.

Because of diminishing pressure and expansion of the potential flow lines, there is a tendency for the flow of water along the run to peel away from the surface of the ship. In fact, this hydrodynamic flow characteristic of *separation* must exist someplace along the run. Preventing separation or delaying it along the hull's surface cannot be effectively done. The magnitude of the eddy currents produced by separation is an indication of the seriousness of the problem in any particular ship form.

Between the entrance and the run in most large commercial ships is a region where the transverse sections below the waterline are constant and identical in shape. This is called the *parallel middle body*. Ships with large parallel middle bodies and high prismatic coefficients (C_p) have relatively high cargo capacities. Ships of very high or extreme block, prismatic, and midsection coefficients are the ore carriers, supertankers, and other bulk-cargo carriers. Ships of lower form coefficients are generally the smaller vessels: working vessels, fishing trawlers, tugs, naval destroyers, and sailing yachts.

Most naval vessels do not have a parallel middle body. Naval combatant ships such as frigates, destroyers, cruisers, and aircraft carriers are characterized by smooth, gentle curvatures and sweeping lines, from a sharp or fine entrance, through a well-rounded, easy run, terminating generally with a transom stern.

The shape of the midsection is characteristic of size and form as well as function. Several such typical sectional forms are shown in figure 3-20. The midsection coefficients indicated are also typical.

3-4 Planing Hulls

Notice that one midsection features a noticeable corner. This is typical of many small craft capable of higher speeds. The corner or edge is the *chine*, and hulls with this characteristic are called chine form. Meant for light displacement and planing operation, a chine-form hull is essentially a multi-surfaced hull, the bottom being distinct from the sides and separated from each by the chine.

Because of their dynamic requirement, planing hulls must all be of a basically similar, characteristic form. The requirements dictate a broad, flat supporting surface to bear and distribute the dynamic supporting pressure. A shallow V-shaped section or dihedral angle in

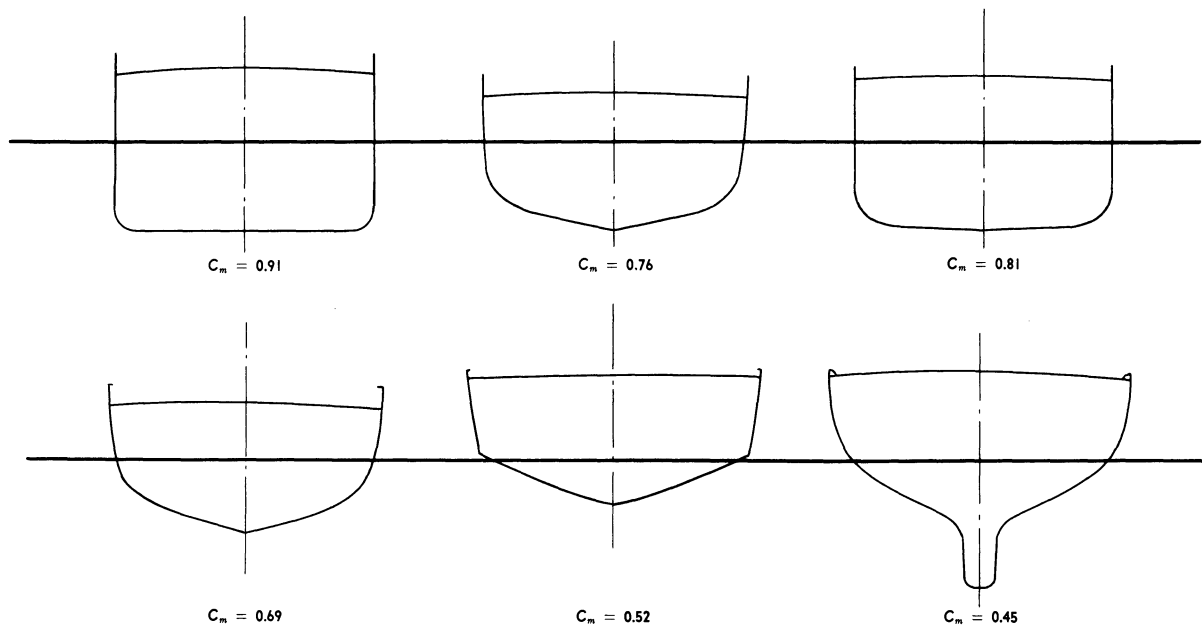


Figure 3-20. Typical midship sections

this surface is necessary to provide adequate directional stability. In general, the major requirements both of high-speed dynamic support and of operating in a seaway at high speed have resulted in a form commonly referred to as a *hard chine* form. The chine, well above the waterline at the bow, curves down to the waterline at about one-third of the waterline length and thence to a very flat angle running out to the lower corners of a broad, flat transom stern (see figure 3-21). Because planing is, to a great extent, dependent upon trim, the longitudinal position of the center of gravity is extremely important. A planing craft operates most efficiently at a planing angle of approximately 3.5° . A small shift in the longitudinal position of G might easily change the trim by this amount. Trim by the bow or the stern in amounts of this magnitude or sometimes less will result generally in a failure to reach planing speed.

The description of the hull form indicated above was intentionally brief. Like displacement hulls, planing hulls must be designed specifically for the requirements demanded of them. The speed range, gross weight, length limitations, and water conditions are the major variables that affect the form of a planing hull. The most satisfactory form of hull to fit any given combination of these variables is determined best by model experiments.

One of the most successful high-speed planing forms incorporates a constant V-section dihedral from the transom to amidships (or slightly aft of it). Horizontal, longitudinal surface ridges are often built into the hull's planing bottom surface. These ridges are spaced closely and run essentially parallel to straight buttock lines along the bottom.

THE SINGLE-CHINE FORM

The most frequently used planing form is the single hard chine described in the preceding section. This form has the advantages of (1) being a substantially good planing surface form, (2) being simple and economical to produce, and (3) having excellent accommodation space for machinery, armament, and crew.

As a planing form, it has the disadvantage of a greater wetted surface when planing than the types outlined below, with consequent greater resistance. Its characteristics in a seaway compared to other planing hulls are only fair, but it must be remembered that all planing hulls are poor in rough water. Altogether, however, this form is probably the most satisfactory type of planing hull for general utility work and military applications when moderately high speeds are required (figure 3-22).

THE STEPPED-CHINE FORM

The stepped-chine form, in which the bottom is made up of two or more planing surfaces in successive steps,

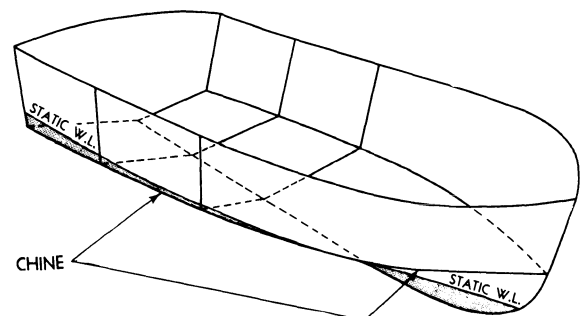


Figure 3-21. Typical hard-chine planing hull form



Figure 3-22. PT 809 from aft (note dry transom). These planing craft are of welded aluminum construction.

is a modification of the hard-chine form. In most such stepped forms, only one step is used, but three and four steps have been used experimentally and in racing hulls.

The single-stepped form has been used in water-based aircraft hulls and has the advantage of quickly reducing wetted surface when accelerating as well as restricting the position of the center of pressure. This form of hull is important where rapid acceleration to maximum speed is desirable. Disadvantageously, the stepped-hull form has even worse rough-water characteristics than the simple-chine form because of the disproportionate distribution of buoyant volume toward the bow. In general, it is capable of higher speeds than other comparable planing forms primarily because of the decreased aspect ratio and decreased wetted surface with consequent decreased induced and frictional drag.

THE INVERTED-V FORM

Planing hulls of the inverted-V form have a characteristic bottom surface, while the normal V bottom surface is inverted with the apex at the top instead of the bottom. This produces sections that form a peaked tunnel longitudinally along the bottom. The apex of the sections flattens as it approaches the stern, and the aftmost sections are essentially flat across the bottom.

The primary advantage of this type of hull is the improved rough-water characteristics caused by the cushioning action of the water in the inverted-V section. There may be slightly improved resistance character-

istics produced by a greater aerodynamic lift caused by the tunneling effect under the bottom.

It must be pointed out in summarizing that while planing hulls have the attractive advantage of high speed with a relatively moderate power expenditure, there are specific disadvantages that limit their usefulness in naval service. *Limited variation in operating displacements* places a premium on the amount of fuel capacity (which means cruising range) or the amount of payload. And *poor rough-water characteristics* result in limited time of operation or, if operation is undertaken in rough water, uneconomical powering characteristics at necessarily lower speeds. Modifications in the planing hull form are frequently made to adapt them, to some extent, to rough water. These modifications, while improving the hull's rough-water ability, invariably decrease its effectiveness as a high-speed planing hull in flat water. Such modifications, however, will often preserve high-speed characteristics to the extent that partial planing is possible.

3-5 Origin and Design of Ship Form

With few exceptions, the process of originating the hull form of a new ship remains a unique function for genuine, creative engineering design. There are many procedures in modern engineering referred to as design. It appears that the number of such procedures has multiplied proportionately with the growth of computerized analysis and popular systems concepts. It is possible to arrange, with the help of a computer and available machinery components, an optimized mechanical system. Computers are able to provide a data base of very useful design information. Mathematical analysis and lengthy computations are accomplished easily by computers. Stability problems are solved, as well as powering and seakeeping analyses. All of this is part of the designer's information needed for optimization, but none of it is, in itself, original design. There are so-called design courses in engineering schools that are actually no more than problem analysis courses in which students become equipped to select combinations of ready-made hardware to perform given functions. In the modern world there are certainly many demands for these functions, but performing them should not be confused with creative design.

Ship design has often been described as an art, sometimes in a derogatory way by engineers whose creativity and vision are limited. It is certainly true, as was pointed out earlier in this chapter, that it is not presently practical to create genuine ships' lines mathematically, which limits the computer in originating ship forms. However, the successful naval architect who designs a ship's hull must be an engineer educated in the various engineering sciences with special emphasis on current hydrome-

chanics and structural theories. Beyond this professional knowledge, designing a new ship form involves something that is often neglected in the naval architect's training: the capacity to visualize a form three dimensionally and translate the object to a communicable graphic medium.

The ship designer's first encounter with this problem comes very early in the design process. Because the ship's hull form is the restricting shape that must contain the mechanisms for propulsion, living accommodations, cargo holds, and all of the functional media of ship operation, it must be determined very early. The following general notes on this procedure may be helpful to a young designer. The development of his capacity to cope with the process and thoroughly understand it will most certainly provide him with confidence. His will be the satisfaction of knowing how to function and produce in the most fundamental and important phase of ship design.

THE LINES AND THEIR DRAFTING

In the beginning of the delineation process, it can be assumed that certain dimensional characteristics are already closely established. These are the length, breadth, depth, draft, and volume of displacement.

Form coefficients are helpful in the beginning, and having some knowledge of ship type and speed, the designer is able to choose an approximate prismatic coefficient. At this point he is also able to plot a curve of areas that represents the shape of this prismatic coefficient, as described below.

CURVE OF AREAS

After selecting a suitable prismatic coefficient, it is possible to compute a corresponding midship-section area. On coordinates with suitable and convenient scales, and with section area as ordinate and length-on-waterline as abscissa, the midsection area is plotted. Through this point, a fair curve of slightly bell-shape form is sketched. The area under this sketched curve, when divided by

the area of the circumscribing rectangle, should be the ratio equal to the prismatic coefficient. Such an equality is achieved by reworking the curve and maintaining its fairness until it is satisfactory. Such a curve is never actually symmetrical, since the greater proportion of its area is abaft the midsection. The total area under this curve represents volume of displacement, and its distribution accounts for the characteristic prismatic coefficient. Comparisons with sectional area curves for existing or prior ship designs of the same type should be made at this stage of the process.

When this preliminary sectional area curve has been satisfactorily determined, it is possible to obtain the below-water areas for any section throughout the ship's length. Such areas should be used as guides and approximations but should not be too restrictive in the following processes of fairing out the lines.

THE GRID

With the dimensional features of the ship known and the above areas and coefficients prepared, the first step in the graphic delineation of the hull is the establishment of the reference grid.

This basic layout of the *waterline*, *section*, and *buttock lines* in their *straight line* projection—with the proper spacing in the sheer plan, half-breadth plan, and body plan—must be very precisely done. Any small dimensional error or lack of parallelism at this point dooms the ultimate fairness of the drawing and agreement between views. Some designers place this grid on the reverse side of the drawing vellum on which they are drawing the ship's lines. This will preserve its integrity during the process of routine erasure that must occur while fairing the lines (figure 3-23).

The profile of the ship showing the contours of the stem and stern as well as the sheer line or weather deck at edge should first be drawn in the sheer plan. This, the first evidence of the emerging ship's hull form, is the designer's first look at the ship's image. At this stage, the shape and contours of this plan are arbitrary and independent of other views, and the designer is

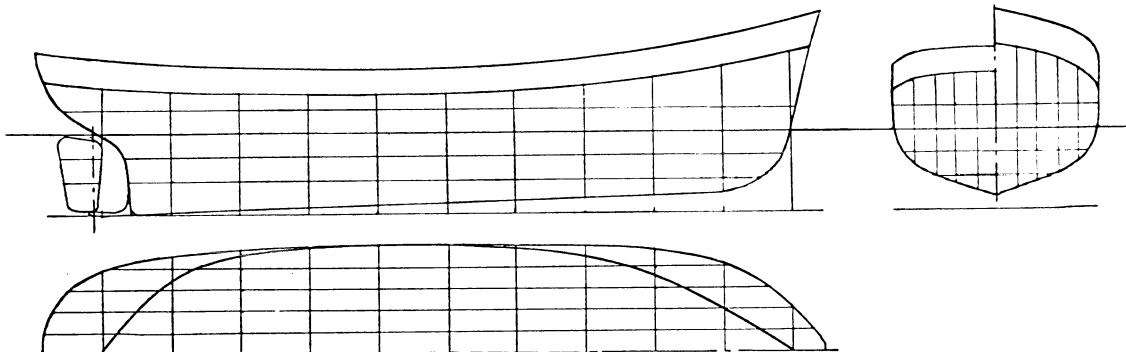


Figure 3-23. Basic grid and profiles: lines development

able to work freely within the limits of the overall criteria.

The next lines or curves should be drawn in the half-breadth plan, and they should be the shape of the designed waterline and the weather-deck edge (sheer in half-breadth). These lines are again fairly arbitrary and freely established. They, too, should reflect the designer's experience and the criteria requirements of the vessel.

The designer next moves to the body plan and, projecting the governing points from the other two views, draws in the midship section. At this point, he must give some regard to the sectional area previously estimated and the midship section coefficient. When he is satisfied with the midsection shape, he returns to the sheer plan. Using the limiting points of intersection, he runs in a bow and buttock line at approximately the quarter-beam distance from the centerline. (This is the quarter-beam buttock.) When this line is faired and its intersections check out in all views, he returns to the body plan and puts in two more sections, preferably numbers two and eight. These must check out in all views before proceeding (figure 3-24).

It becomes obvious at this stage that each additional line produces an accumulating number of intersections that must agree in all views for fairness. The process of working in a sequence of views and reference planes rather than attempting to draw all or several waterlines, all the buttock lines, or all sections is important. The shape is being developed by gradually "caging" it in—by building an integrated structure of limits of cross-referenced intersecting planes.

The sequence should continue, with buttock line, waterline, and section, until all of the lines are completed. Because the buttock lines are involved with only half the ship (from the center plane of symmetry), there are less of these than the sections and waterlines. Hence, the sequential procedure concentrates primarily on waterlines and sections, and interspersing buttock lines when necessary.

Finally, as a check on fairness when the lines are nearly completed, a fourth reference cutting plane called

a *diagonal* is established. Depending upon the designer's requirements and the size and shape of the ship, one or several diagonals may be used. Larger vessels often need but one. The diagonal plane's position is established in the body plan as an edge-viewed plane, at an angle to the centerline plane passing downward through the hull near the bilge, in an attempt to intersect the hull surface as nearly perpendicularly as possible.

The projection of this intersection after rotating it parallel to a basic reference view, generally the half-breadth plan, is called the *development of the diagonal* and is shown as such in figure 3-3.

TABLE OF OFFSETS

When the lines are completed and faired (all intersections in agreement in three views) and the designer is satisfied with the hull form, a rectilinear tabulation of the reference plane intersections is made. The intersections, normally tabulated according to the section or station numbers, include the heights above the base line of each section, where it is crossed by a bow and buttock line, as well as the deck edge and stem and stern contour. Also tabulated are the half-breadths at each section where they are crossed by the waterlines. The referenced distances are recorded in feet, inches, and eighths of inches (such as 6-4-3, meaning 6 feet 4 $\frac{3}{8}$ inches) full ship scale, or in decimal format.

This first table of offsets is often referred to as *uncorrected offsets* and is sent along to the builder with the ship's plans. In the past the loftsmen in the shipyard's mold loft laid down the lines from the offsets on a large wooden floor (loft), where the enlargement revealed small discrepancies undetected in the designer's scale drawing. The offset table was corrected to this larger fairing process, and a corrected or lofted offset table was made. This process is largely obsolete except in small shipyards, and the fairing process is now accomplished by computers.

Further use is made of the offset table as computer input, not only in the fairing process but also in the many tedious calculations made for hydrostatic char-

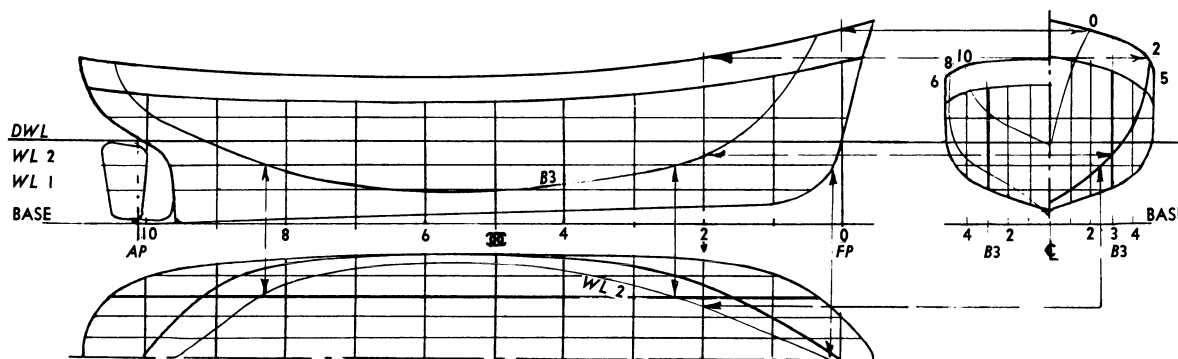


Figure 3-24. Development of lines

acteristics and stability. In modern ship design, it becomes the basis of the ship's configuration for further computer modification and refinement toward the preliminary and contract design stages.

3-6 Hydrostatic Parameters

Over the years the ship designer, in order to facilitate his computations and articulate his expression of ship characteristics, has developed a number of classic parameters. The basis of these expressions, as well as their method of application, must be understood by the student of this profession. Before discussing these significant expressions, it will be useful to restate Archimedes' Principle as it applies to surface ships, for it is from this expression that the hydrostatic parameters have grown: *The total weight of the floating vessel, including all that it contains or that is attached, equals the weight of the water it displaces.*

A further statement of this law is as follows: *A body immersed in a fluid will experience an upward force due to hydrostatic pressure equal and opposite to the weight of the fluid displaced by the body.* With this statement, we have the ultimate substantiation for the phenomenon that applies to submersible craft, submerged objects, or appendages, regardless of the "floatability" of the vessel.

WATER DENSITY AND DISPLACEMENT

The displacement of a ship at any draft is obtained by dividing the submerged volume at that draft expressed in cubic feet by the appropriate density factor (35 cubic feet per ton of salt water, or 36 cubic feet per ton of fresh water). Using the submerged volume at the designer's load waterline, the *designed displacement* may be computed. From these calculations, a curve of displacement versus draft can be drawn, referred to as the *displacement curve*. It will be discussed further.

TONS-PER-INCH IMMERSION

When a ship undergoes a change in displacement, it is extremely useful at times to make a quick computation of the change in draft. The most convenient and rapid method is to divide the change in displacement (in tons) by the *tons-per-inch immersion (TPI)*. This latter factor is exactly what its name implies: the number of tons required to produce a change in mean draft of one inch in salt water.

The sides of a ship over a vertical distance of one inch are nearly vertical at most operating drafts. This means that there will be no noticeable change in the area of the waterplanes in a change of draft of one inch. If we can make this assumption, then it is apparent that the additional volume displaced by the ship in sinking one inch will be the product of the waterplane area in square feet and the thickness of the layer in feet ($\frac{1}{12}$).

The weight in tons of the increased volume is $\frac{1}{35}$ of the volume. Hence,

$$\frac{\text{area of waterline}}{35} \times \frac{1}{12} = \text{tons-per-inch immersion (TPI)}$$

or simplified

$$\text{TPI} = \frac{A_{wp}}{420} \quad (3-15)$$

To recapitulate, the above equation indicates that the weight of a layer of displaced water is equal to the area of the waterplane times a thickness of one inch divided by 35, the volume in cubic feet of one ton of seawater. This change in displacement, according to Archimedes' Principle, must be exactly equal to the change in weight that causes it.

The ship's curves of form, which include displacement, give the value of tons-per-inch immersion in salt water for the complete range of drafts. Unless otherwise specified, TPI is *always* given for salt water.

In using the value of TPI, the student must remember the original assumption that the sides of the ship are nearly vertical throughout the change of drafts for which TPI is used. This means simply that for large changes in drafts the value may be considerably in error. In such case, a draft somewhere midway between the original and the final drafts would give a more correct value for TPI when TPI is obtained from the curves. In any case, common sense must be the guide in working with TPI, keeping in mind the form of the ship and the range between the initial and final drafts. If it becomes necessary to use TPI in fresh water, the following equation applies:

$$\text{TPI}_{fw} = \frac{35}{36} \text{TPI}$$

DRAFT-VERSUS-DENSITY RELATIONS

Because of the variable densities between salt and fresh water, a ship of a given weight or displacement will have different drafts in the two liquids.

It is useful to predict the change in draft when moving from salt to fresh water, or from fresh to salt. Many students are inclined to make the hasty generalization that if the ratio of densities of fresh and salt water is $\frac{35}{36}$, then they have merely to multiply the draft in fresh water by this ratio to obtain the draft in salt water. This is not correct in the case of ships, for a moment's inspection will reveal that the density ratio is a function of volumes, whereas the draft is only a single linear dimension of three variable dimensions making up the volume of the ship. Consider for a moment two ships of equal displacement at the *same drafts*. One ship is wide and full with a flat bottom, while the other is sharp ended and narrow with considerable deadrise. Consider

both of these ships passing from fresh water to salt water simultaneously. The change in draft caused by the change in water density reflects the change in buoyant layer. Archimedes' Principle explains why the ships are buoyed up by water of a greater density than the original and hence will rise in the water until the weight of the volume of water displaced equals their own weight. In the case of the wide, full ship, the displacement volume is more a function of her fullness in sides and ends and hence requires less change in draft to make up the change in buoyant volume. In the case of the ship with a sharper, narrower hull, a greater change in draft must result to compensate for the lack of fullness in the hull.

The correct approach to this situation may be developed as follows: A ship is passing from water of specific gravity γ to water of specific gravity γ_1 . The displacement Δ is the same in both cases, but the volume of displacement changes.

$$\begin{aligned}\nabla &= \text{volume before change} = \frac{36\Delta}{\gamma} \\ \nabla_1 &= \text{volume after change} = \frac{36\Delta}{\gamma_1}\end{aligned}$$

The change in volume, ∇_1 minus ∇ , can be considered a layer of uniform thickness of volume $A_w \delta T$ in cubic feet, where

A_w = waterplane area in square feet

δT = thickness of layer in feet.

The distance δT is the change in draft in passing from water of one density to another, assuming A_w remains constant throughout the distance δT , which is a close approximation.

Therefore

$$\begin{aligned}A_w \delta T &= \nabla_1 - \nabla \\ \delta T &= \frac{1}{A_w} (\nabla_1 - \nabla)\end{aligned}$$

In terms of the displacement from above,

$$\delta T = \frac{36\Delta}{A_w} \left[\frac{1}{\gamma_1} - \frac{1}{\gamma} \right]$$

The above equation, expressed in terms of TPI (tons-per-inch immersion), where

$$A_w = 35(12\text{TPI})$$

is

$$12 \delta T = \frac{36\Delta}{35\text{TPI}} \left[\frac{1}{\gamma_1} - \frac{1}{\gamma} \right] \quad (3-16)$$

Note that the sign of the numerical value of δT indicates whether change is an increase or decrease in draft. In

the case of a ship passing from *salt to fresh water*, the formula reduces to

$$\begin{aligned}12 \delta T &= \frac{36\Delta}{35\text{TPI}} \left(1 - \frac{35}{36} \right) \\ 12 \delta T &= \frac{\Delta}{35\text{TPI}}\end{aligned} \quad (3-17)$$

In the case of a ship passing from *fresh to salt water*, the formula becomes

$$12 \delta T = - \frac{\Delta}{35\text{TPI}} \quad (3-18)$$

Equations 3-16, 3-17, and 3-18 all give the change in draft in inches.

3-7 Hydrostatic Curves (Curves of Form)

The large folded print in the pocket in the back of the book is an exact copy of the original *curves of form*, prepared by the designers for the destroyer USS *Gearing*. (The DD 692 class is typical of all other ships of similar size and seagoing purpose and will be used henceforth for examples throughout this book.) These curves are typical and represent the standard presentation of hull characteristics that are functions of form. The most convenient way of recording these characteristics graphically is on coordinates of mean drafts versus displacements. Because the basic variation in underwater hull form is the result of varying drafts, the ordinate scale is in *feet of mean draft*. The abscissa is in *tons of displacement*. For functions of drafts other than displacements, suitable scale factors are provided to convert the reading in tons to the proper dimensions.

The following description of the individual curves will serve as a guide to their use and preparation:

Curve 1—Displacement in salt water (Δ_{sw}). The draft scale used for this and all subsequent curves is the mean draft to the bottom of the keel. This is below the molded base line, a distance of the thickness of the keel. It is sometimes referred to as the *base line at the bottom of the keel amidships*.

The salt-water displacement curve is probably the most frequently used of all the curves, because it is generally the starting point for the use of the other curves in obtaining other data.

The method of using this curve is practically self-explanatory. If the displacement is desired, enter the ordinate scale with the mean draft, note the intersection of the horizontal line corresponding to the given mean draft with curve 1, and read the corresponding displacement in tons on the horizontal scale vertically above. If the displacement is given and the mean draft is desired, the procedure is simply reversed.

Curve 2—Displacement in fresh water (Δ_{fw}). For the same mean draft, the ship's displacement will be less

in fresh water than in salt water, or conversely, for the same displacement or weight (the more logical case), the ship will float at a lesser draft in salt water than in fresh water. The horizontal separation of curves 1 and 2 is proportional to $^{35/36}$, the density ratio of pure fresh water and standard seawater. Curve 2 is used in the same manner as curve 1.

Curve 3—Center of buoyancy above bottom of keel amidships ($\overline{KB}$). The use of this curve involves translating a displacement or draft reference into a linear value of $\overline{KB}$. This is done, as in the case of most of the subsequent curves, by the use of a scale factor. For this curve, as for others, the scale factor is printed on the curve itself. For $\overline{KB}$, it is 100 tons equal 1 foot. The value is read by entering with the mean draft, reading horizontally to curve 3, and thence vertically to the displacement value above. The reading is converted to feet by the scale factor given above.

Curve 4—Center of buoyancy aft of station 10 (longitudinal position of the center of buoyancy [LCB]). This curve gives the longitudinal location of the center of buoyancy B with reference to the midship section, in this case section 10. (This section is sometimes called the *midship perpendicular* [MP].) Enter with the given mean draft and read to the displacement scale vertically above. This reading, as above, is divided by the scale factor 100 tons equals 2 feet, which gives the longitudinal distance of B from section 10 in feet.

Curve 5—Area of waterlines (A_w). This curve provides the area in square feet of any waterplane parallel to the base plane at a given mean draft. Entry and reading are made in the same way as for curves 3 and 4 above. The scale factor in this case is 100 tons equal 1,000 square feet.

Curve 6—Center of gravity of waterplane aft of section 10 (LCF). This curve provides the longitudinal location of the center of flotation with reference to the midship section. By definition, the center of flotation is the centroid or geometric center of the waterplane. The distance of this point, as given from section 10, may be reoriented for convenience in trim problems to the forward perpendicular by simply adding half of the length between perpendiculars. Entry and reading are made as before, using the scale factor 100 tons equal 2 feet.

Curve 7—Tons-per-inch immersion (TPI). This curve provides a source for values of TPI at any mean draft without recourse to the computation described in the foregoing section. The values, as defined for this characteristic, are given for *salt water only*. If fresh water values are desired, the plotted values must be multiplied by $^{35/36}$. Entry and reading are made, as before, using a scale factor of 100 tons equal 2 tons per inch.

Curve 8—Area of station 10 (midship section), (A_M). This curve gives the value of the area of section 10 up to any given mean draft. It is one of the more important

hull characteristics and is provided on all typical curves of form. The values are obtained similarly to those for curve 7 using the scale factor of 100 tons equal 50 square feet.

Curve 9—Station 10 with plating (outline of the midship section). This curve provides essentially what its name implies. Its coordinates are to the same scale as is the draft scale in feet. It shows the actual shape of the midship section to the outside of the plating for the half-breadth of the ship. The scale factor is 100 tons equal 1 foot.

Curve 10—Transverse metacenter above bottom of keel amidships ($\overline{KM}$). This curve provides the frequently used value of $\overline{KM}$ for the complete ranges of mean drafts encountered. Entry and reading is made in the usual manner. The scale factor is 100 tons equal 2 feet. Note that since $\overline{KG}$ is not determined by ship geometry alone, the metacentric height $\overline{GM}$ cannot be included in the curves of form.

Curve 11—Longitudinal metacentric radius ($\overline{BM}_L$). The longitudinal metacentric radius must not be confused with the transverse metacentric radius (the value that is more frequently used). Values are obtained as before, using the scale factor 100 tons equals 200 feet.

Curve 12—Approximate moment to alter trim one inch (MTI''). The values given here are the moments to change trim one inch for any of the various mean drafts involved. The values are labeled *approximate*, inasmuch as they are obtained from the approximate relationship

$$\frac{\Delta \times \overline{BM}_L}{12L} \quad \text{rather than the exact relationship}$$

$$\frac{\Delta \times \overline{GM}_L}{12L}. \quad \text{The values are obtained in the usual man-}$$

ner, using a scale factor of 100 tons equal 100 foot tons per inch.

Curve 13—Addition to displacement due to one foot change of trim by stern. A vessel that is trimmed by the stern at any given mean draft generally has a greater displacement than the same untrimmed ship at the same mean draft. This is the result of the conventional ship forms that have increasingly fuller waterlines aft as draft increases. The converse is true when the ship is trimmed by the bow.

Although the correction to the untrimmed displacement is small unless the trim is excessive, it is a factor that must be applied for an accurate value of displacement.

A ship generally has a greater immersed volume when trimmed by the stern than when floating on an even keel at the same mean draft as given by curves 1 or 2. The correction to be added to the displacement for one foot of trim by the stern is taken from curve 13, using scale factor 100 tons equal 2 tons per foot. For a trim of other than one foot, correct the value obtained in

proportion to the amount of trim. When the ship is trimmed by the bow, the correction is subtracted.

Curve 14—Area of the wetted surface (S). The wetted surface is used in making resistance calculations in chapter 11.

Curve 15—Section area curve. The curve of section areas up to the loaded waterline represents the distribution of underwater volume along the length of the ship.

Curve 16—Outboard profile. The outboard profile shows the relative location of frames and stations, including frame and station spacing and the length between perpendiculars L_{PP} .

Example 3-1: Information is available from the curves of form. The following questions apply to the USS *Gearing*, DD 710 (692 class), which is at sea with a draft forward of 14 feet 6 inches and a draft aft of 13 feet 6 inches.

a) What is her mean draft?

$$T_m = \frac{T_F + T_A}{2} = \frac{14'6'' + 13'6''}{2} = \boxed{14 \text{ feet}}$$

b) What is her displacement in salt water?

$$\Delta (\text{curve 1}) = 3,350 \text{ tons at level trim}$$

$$\delta\Delta (\text{curve 13}) = 995 \times \frac{2}{100} = 19.9 \text{ tons/ft trim}$$

$$\Delta = 3,350 + \frac{-1}{1} (19.9) = \boxed{3,330 \text{ tons}}$$

c) Calculate the block coefficient at level trim.

$$B (\text{curve 9}) = 2,030 \times \frac{2}{100} \times 2 = 40.6 \text{ ft}$$

$$\begin{aligned} C_B &= \frac{\Delta 35}{LBT} \\ &= \frac{3,350 \times 35}{383(40.6)14} \\ &= \boxed{0.54} \end{aligned}$$

d) Calculate the midships section coefficient.

$$A_M (\text{curve 8}) = 950 \times \frac{50}{100} = 475 \text{ ft}^2$$

$$\begin{aligned} C_M &= \frac{A_M}{BT} \\ &= \frac{475}{40.6(14)} \\ &= \boxed{0.84} \end{aligned}$$

e) Calculate the waterplane coefficient.

$$A_w (\text{curve 5}) = 1,190 \times \frac{1,000}{100} = 11,900 \text{ ft}^2$$

$$\begin{aligned} C_w &= \frac{A_w}{LB} \\ &= \frac{11,900}{383(40.6)} \\ &= \boxed{0.77} \end{aligned}$$

f) Determine the location of the LCF , $\overline{XF}$.

$$\begin{aligned} \overline{XF} (\text{curve 6}) &= 1,120 \times \frac{2}{100} \\ &= \boxed{22.4 \text{ feet}} \text{ aft of } \overline{X} \end{aligned}$$

g) Determine $\overline{GM}$ if $\overline{KG} = 15.5$ feet.

$$\begin{aligned} \overline{KM} (\text{curve 10}) &= 960 \times \frac{2}{100} = 19.2 \text{ ft} \\ \overline{GM} &= \overline{KM} - \overline{KG} \\ &= 19.2 - 15.5 \\ &= \boxed{3.7 \text{ feet}} \end{aligned}$$

h) Determine the height of the longitudinal metacenter above the keel.

$$\begin{aligned} \overline{KB} (\text{curve 3}) &= 840 \times \frac{1}{100} = 8.4 \text{ ft} \\ \overline{BM}_L (\text{curve 11}) &= 435 \times \frac{200}{100} = 870 \text{ ft} \\ \overline{KM}_L &= \overline{KB} + \overline{BM}_L \\ &= 8.4 + 870 \\ &= \boxed{878 \text{ feet}} \end{aligned}$$

i) After a period of steaming, the mean draft decreases by two inches at the same trim. How many tons of fuel and stores have been consumed?

$$\text{TPI} (\text{curve 7}) = 1,420 \times \frac{2}{100} = 28.4 \text{ tons/in}$$

$$\begin{aligned} \delta\Delta &= 2 \text{ in} \times \text{TPI} \\ &= 2(28.4) \\ &= \boxed{56.8 \text{ tons}} \end{aligned}$$

Problems

3-1. Calculate the prismatic coefficient and the transverse waterplane inertia coefficient for the ship in example 3-1.

3-2. A barge with vertical sides has the shape shown

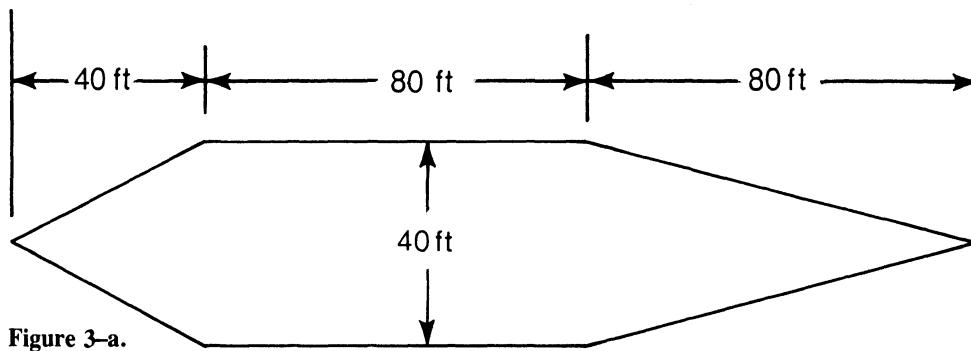


Figure 3-a.

in figure 3-a when viewed from above. Determine the following characteristics of the barge and compare them with a rectangular barge of the same length, beam, and draft:

- Displacement at a draft of 15 feet
- Block coefficient C_B
- Waterplane area coefficient C_{WP}
- Centroid of the waterplane area (longitudinal center of flotation)
- Transverse waterplane inertia coefficient C_{IT}
- Longitudinal waterplane inertia coefficient C_{IL} about the centroid of the waterplane area

3-3. The following questions apply to the USS *Gearing*, DD 710, which is at sea with a draft forward of 14 feet 8 inches and a draft aft of 13 feet 4 inches:

- What is her mean draft?
- What is her trim?
- What is her tons-per-inch immersion?
- What is her displacement in salt water?
- What is her transverse metacentric radius $\overline{BM}$?
- Determine her new mean draft if she steams from salt water to fresh water.

3-4. The following questions apply to the USS *Gearing*, DD 710 (692 class), which is at sea with a draft forward of 13 feet 5 inches and a draft aft of 12 feet 1 inch:

- What is her mean draft?
- What is her displacement in salt water (tons)?
- If her beam at this draft is 40.4 feet, what is her block coefficient at level trim?
- What is her transverse metacentric radius $\overline{BM}$?
- After a period of steaming, the mean draft decreases by 2 inches at the same trim. How many tons of fuel and stores have been consumed?

3-5. The following questions apply to the USS *Gearing*, DD 710 (692 class), which is at sea with a draft forward of 12 feet 10 inches and a draft aft of 14 feet 8 inches:

- What is her mean draft?
- What is her trim?
- What is her tons-per-inch immersion?
- What is her displacement in salt water (tons)?
- What volume of seawater is displaced at this draft (ft^3)?
- If her beam at this draft is 40.4 feet, what is her block coefficient at level trim?
- What is her waterline coefficient?
- After a period of steaming, the draft forward changes to 12 feet 7 inches and the draft aft to 14 feet 5 inches. How many tons of fuel have been consumed?

Properties of Shipbuilding Materials

4-1 Stress and Strain

In chapter 2 the concepts of external forces and moments on rigid bodies were discussed. Since all real objects are deformable when subjected to either external or internal forces and moments, methods must be developed to analyze these forces and the resulting deformations. This involves a knowledge of material properties, covered in this chapter, and a knowledge of the mechanics of deformable bodies (solid mechanics) covered in the next chapter. Common to both topics is the concept of stress and strain.

STRESS

Stress is defined as a measure of internal force per unit area. Stresses are normally resolved into *normal stresses*, which have lines of action perpendicular to the area under consideration, and *shear (tangential) stresses*, which have lines of action coplanar with the area under consideration. Normal stresses produced by tensile forces pulling away from the plane area are called *tensile stresses*, and those produced by compressive forces acting towards the plane area are called *compressive stresses*. Shear stresses are produced by force components that act parallel to the plane area under consideration. Since the local force per unit area is seldom constant over a large area, stress must be defined on the basis of a differential volume $\delta x \delta y \delta z$, which is a free-body diagram of an interior section of a material, as illustrated in figure 4-1. The differential force δF acts only on the plane $\delta A = \delta y \delta z$. Additional forces and the resulting stresses act on the other sides of the differential volume.

In setting up the differential element within a material, the assumption must be made that it consists of continuous media rather than isolated atoms the size of the "infinitesimal element" chosen. This is the basis for studying the mechanics of continuous media (continuum mechanics), of which solid mechanics and fluid

mechanics are both examples. In essence, mathematical definitions involve taking limits not to zero but to a very small area or volume that still contains enough atoms to make the definition valid.

The normal stress on a differential area δA_x , with x as a constant and tensile stress considered positive, is defined as

$$\sigma_x = \lim_{\delta A_x \rightarrow 0} \frac{\delta F_x}{\delta A_x} = \frac{dF_x}{dA_x} \quad (4-1)$$

The shear stresses are present in pairs on each of the surfaces of the differential element. In the example given, the first subscript indicates the plane is at constant x and the second subscript indicates the direction of the stress vector.

$$\tau_{xy} = \lim_{\delta A_{x \rightarrow 0}} \frac{\delta F_y}{\delta A_x} = \frac{dF_y}{dA_x} \quad (4-2)$$

$$\tau_{xz} = \lim_{\delta A_{x \rightarrow 0}} \frac{\delta F_z}{\delta A_x} = \frac{dF_z}{dA_x} \quad (4-3)$$

Since three independent stresses exist on each of the three independent planes, the array of nine independent stresses is called the *stress tensor* and is treated in studies of solid and fluid mechanics. Three-dimensional problems will not be treated in this text, however, and the analysis of the stresses will be limited to two-dimensional problems.

Figure 4-2 represents the stresses on a two-dimensional element surface of unit depth. The additional force per unit volume Y represents a body force that acts on the whole element, such as that caused by gravitational or electromagnetic effects. Except for gravity these forces, along with interactions between the stresses, are considered to be higher-order effects and will be neglected in this treatment. The interested reader is

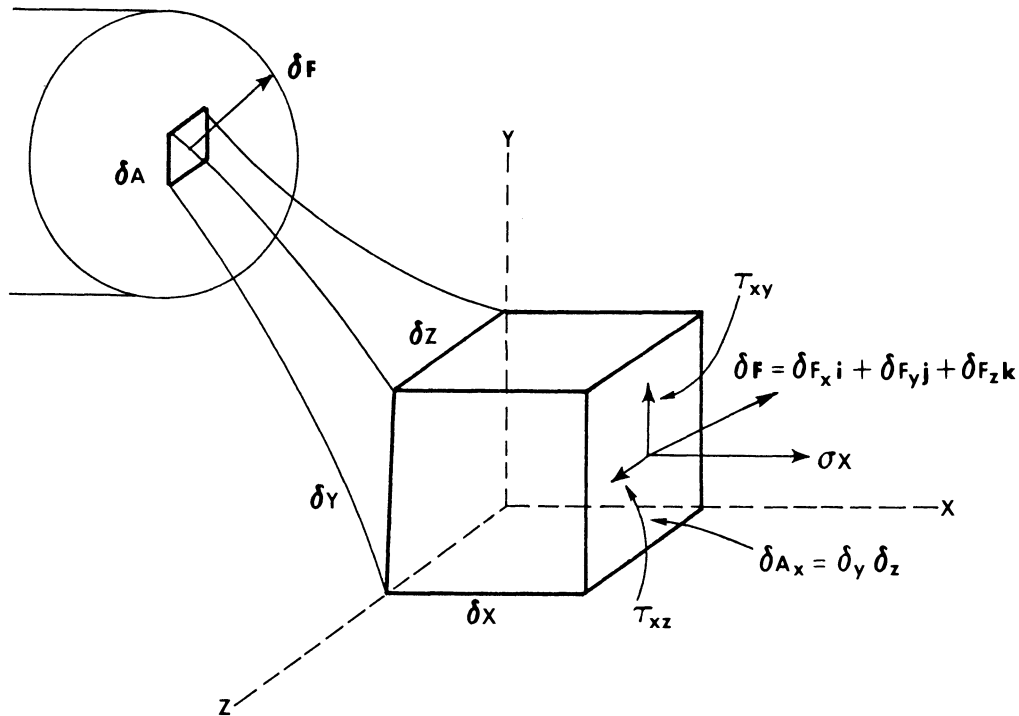


Figure 4-1. Stresses on a differential volume

referred to a textbook on strength of materials for a rigorous analysis of the problem (for example see Popov 1968, chapter 3, and Higdon et al. 1976, chapter 1).

If the higher-order effects just listed are assumed to be negligible on an elemental volume, the normal stresses exist in colinear pairs that are equal in magnitude and opposite in direction for static equilibrium. Likewise, the shear stresses exist in pairs of coplanar couples that must balance out to zero net moment on the differential element; i.e.,

$$|\tau_{yx}| = |\tau_{xy}| \quad (4-4)$$

For nonuniform loading along any axis, the higher order analysis allows the variation in both normal and shear stresses to be calculated for a finite structure.

Uniform Stresses Caused by Axial Loads. Many practical problems such as the application of tensile or compressive loads to a uniform beam or rod allow applied force to be considered uniform over the area of the beam or rod. For these cases the definition of normal (axial) stress is the familiar

$$\sigma = \frac{P}{A} \quad (4-5)$$

where P = the applied axial load

A = the cross-sectional area of the body (figure 4-3).

An average shearing stress may be defined for situations where an axial load is transmitted from one member to another through a "shear pin," rivet, glued

joint, or other means of attachment in which the stressed area is parallel to the direction of the applied load. For the example of the pinned joint in figure 4-4, the actual shear stress varies over the cross-sectional area of the pin, but an average shear stress may be defined as follows:

$$\tau = \frac{P}{A} \quad (4-6)$$

where A = the total cross-sectional area resisting the applied shearing load P .

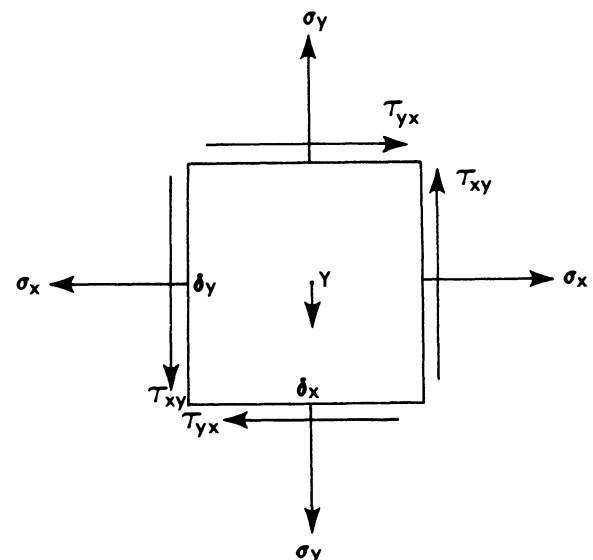


Figure 4-2. Stresses on a two-dimensional surface element

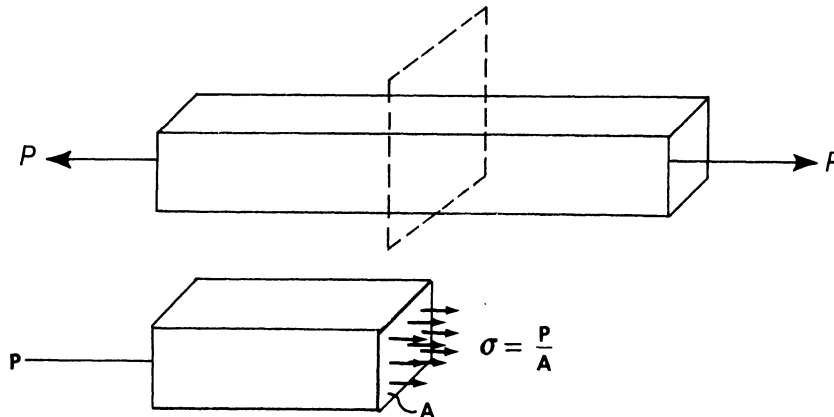


Figure 4-3. Uniform axial stress

The maximum local shear stress may significantly exceed the average value found from equation 4-6.

STRAIN

Strain is defined as a measure of the deformation of a body as a result of the action of external and internal forces (stresses). The deformation can take several forms, such as changes in length (elongations and contractions) and both angular and volumetric distortions. For solids and viscous substances (fluids) of constant density, elongations in one direction are accompanied by contractions in another direction, since the volume distortion of “incompressible” materials is very small. (Compressible fluids of variable density can undergo large volume distortions.)

The analysis of three-dimensional strain involves the use of partial derivatives to set up a *strain tensor* corresponding to the stress tensor discussed previously. However, axial strain caused by “pure” axial loads and shear strain caused by “pure” shearing loads will be analyzed individually.

Axial strain is defined as the axial deformation per unit length of the material. To illustrate this concept, consider a rectangular bar subjected to an axial load P (figure 4-5). Since the bar is uniform in cross section and the pure axial load produces a uniform normal stress in the bar, the axial strain may be defined based on the change in overall dimensions of the bar.

$$\epsilon_x = \frac{e_x}{L} \quad (4-7)$$

If the bar is not uniform in cross section, as illustrated in figure 4-6, then the load per unit area (normal stress) will vary along the bar. Under these circumstances, one would expect the deformation per unit length (strain) to also vary along the length of the bar. To analyze this problem, take a section through the bar at $A-A'$ and analyze the strain on a free-body diagram of a small differential element in the interior of the body, as illustrated in figure 4-7. As the load is applied, the right-hand face of the small element will stretch further than the left-hand face by an amount δe_x . Likewise, the horizontal faces of the element will move toward each other by an amount δe_y and the constant z -faces (not illus-

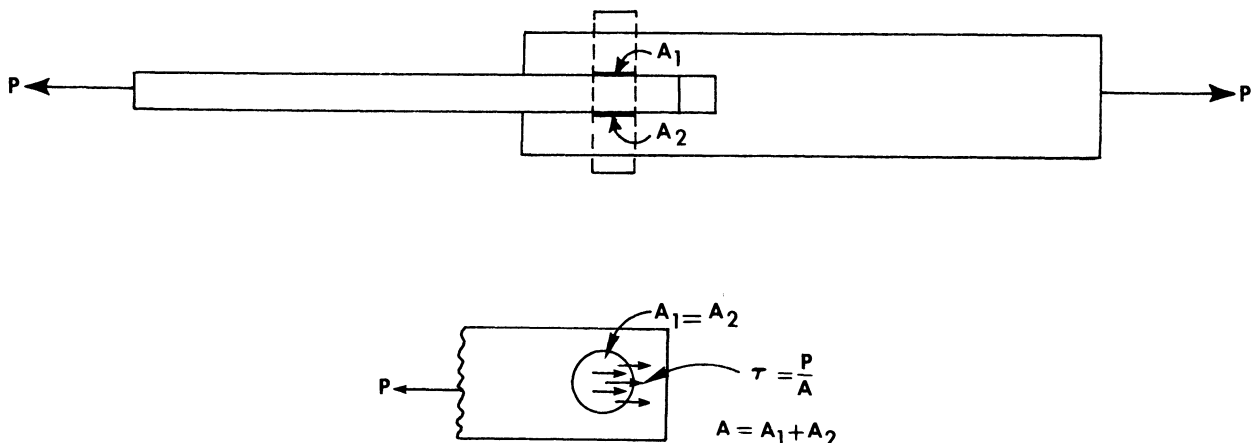


Figure 4-4. Pinned joint in shear

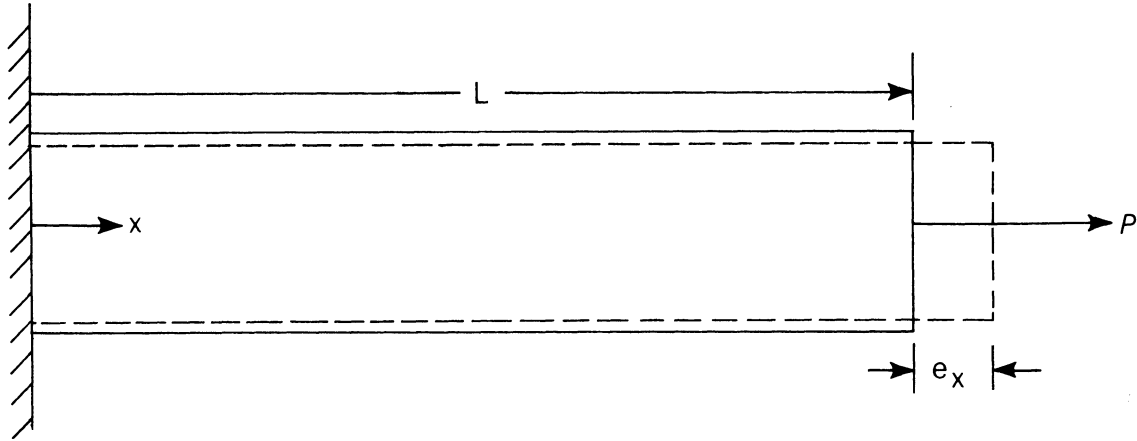


Figure 4-5. Uniform axial strain

trated) would move toward each other by δe_z . The local linear strains may be defined by

$$\epsilon_x = \lim_{\delta x \rightarrow 0} \frac{\delta e_x}{\delta x} = \frac{de_x}{dx} \quad (4-8a)$$

$$\epsilon_y = \lim_{\delta y \rightarrow 0} \frac{\delta e_y}{\delta y} = \frac{\partial e_y}{\partial y} \quad (4-8b)$$

$$\epsilon_z = \lim_{\delta z \rightarrow 0} \frac{\delta e_z}{\delta z} = \frac{\partial e_z}{\partial z} \quad (4-8c)$$

where the partial derivatives indicate that the strains in the y - and z -directions also vary with the distance along the bar. For three-dimensional problems, all the linear strains will be represented by partial derivatives. The customary sign convention considers extensions as positive strains and contractions as negative strains.

Pure *shearing strain* is less common than pure axial strain but may be considered to exist in isolated sections of a shear pin, such as in the areas A_1 and A_2 in figure 4-4. A free-body diagram of a differential element of

that material is shown in figure 4-8. The shear strain γ is defined as

$$\gamma = \lim_{\delta y \rightarrow 0} \frac{\delta u}{\delta y} = \frac{du}{dy} \quad (4-9)$$

In general, all the axes will undergo angular deformations. Two of the deformations are illustrated in figure 4-9. Since δu , δv , and δw in the z -direction will vary in all three directions, partial derivatives must be used in the definitions of angular strain.

$$\gamma_{yx} = \gamma_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \quad (4-10a)$$

$$\gamma_{zx} = \gamma_{xz} = \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \quad (4-10b)$$

$$\gamma_{zy} = \gamma_{yz} = \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \quad (4-10c)$$

These shearing strains can be combined with the axial strains to form a strain tensor, which is utilized in three-dimensional problems.

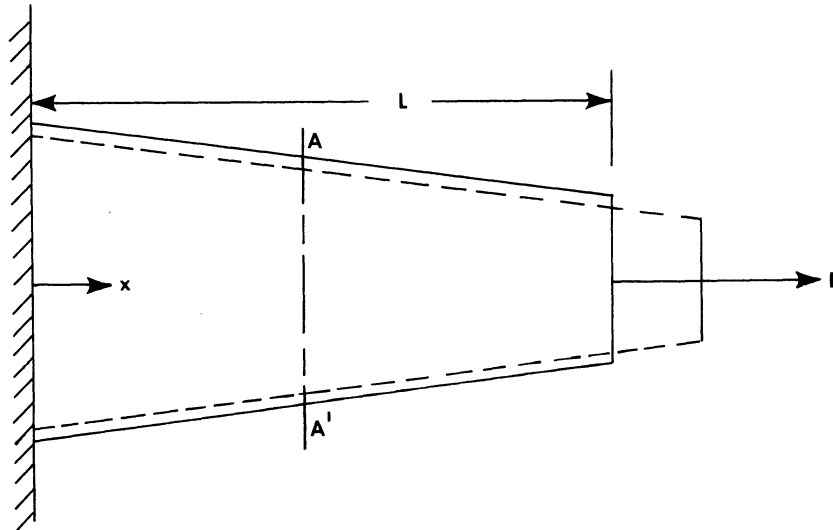


Figure 4-6. Nonuniform axial strain

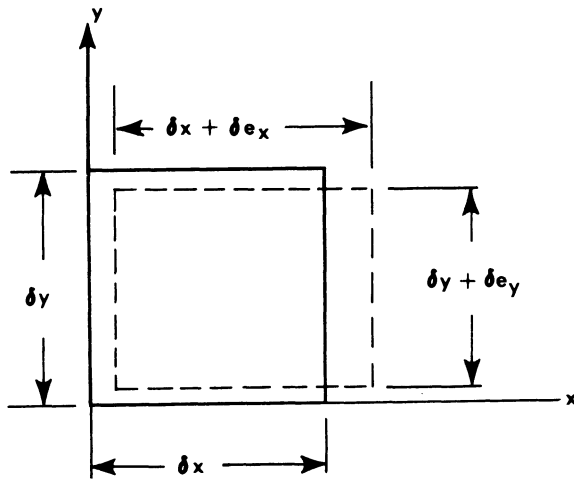


Figure 4-7. Interior element at section AA'

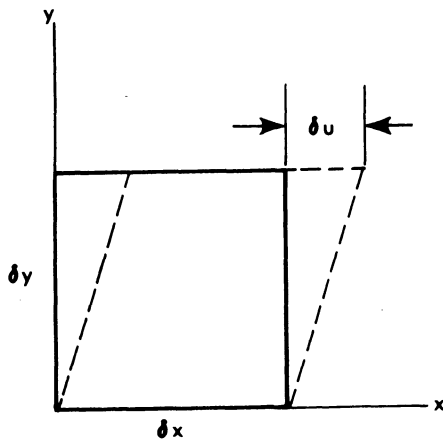


Figure 4-8. Pure shearing strain

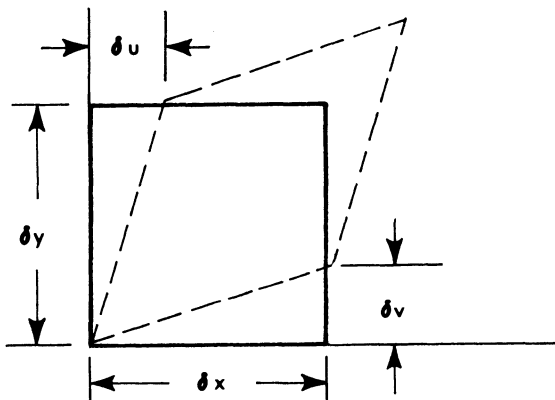


Figure 4-9. Shearing strain

4-2 Elastic and Plastic Deformation: The Relationship Between Stress and Strain

Most solid substances return to their original shape when an applied load is removed. This behavior is called elasticity or elastic deformation. If a critical value of stress known as the *elastic limit* is exceeded, however, the substance will deform permanently. The permanent deformation that remains after the load is removed is called *plastic deformation*. If the plastic deformation is not carried to fracture, the material still exhibits elastic properties.

The elastic-plastic behavior of materials is best illustrated on a stress-strain diagram (figure 4-10). The relationship between tensile stress and axial strain as measured on a tensile testing machine is shown for several types of materials. Rubber-like materials exhibit nonlinear elastic behavior almost to the point of rupture. They are characterized by low values of stress for relatively large values of strain until they stiffen just before failure. Ductile materials such as copper and lead exhibit extreme plastic deformation and have elastic limits that are quite low. Stiff materials such as mild steel may exhibit complex behavior with several important characteristics, defined as follows:

1. *Proportional limit*—The maximum stress for which stress and strain are proportional. It is difficult to determine this limit experimentally since the initial deviation from linear behavior may be quite small.

2. *Elastic limit*—The maximum stress for which the loading and unloading curves follow the same line. It is generally very close to the proportional limit for stiff materials.

3. *Yield point*—The stress at which some materials exhibit a large increase in strain at essentially constant stress. Some steels have multiple yield points with upper and lower values.

4. *Yield strength*—The stress that will induce a specified permanent set. It is defined for those materials that do not exhibit a yield point. It is determined by the intersection of the stress-strain curve with a line drawn parallel to the initial slope of the curve at a 0.2 percent offset in strain. Sometimes other offsets from 0.05 to 0.3 percent are specified.

5. *Ultimate tensile strength (UTS)*—The maximum tensile stress a material specimen will support based on the original cross-sectional area of the material. Since a ductile material generally “necks down” just before breaking, the true tensile stress present in ductile materials will be considerably larger than the UTS based on the original cross-sectional area. Necking down does not occur in brittle materials such as cast iron.

Figure 4-11 shows some typical stress-strain curves for representative ship-hull materials.

HOOKE'S LAW, YOUNG'S MODULUS

In 1678 the English scientist Robert Hooke first observed that the load was proportional to the deformation (stretch) in springs. In 1807 another Englishman, Thomas Young, proposed that the constant of proportionality between stress and strain be called the *modulus of elasticity* (E); i.e.,

$$E = \frac{\sigma}{\epsilon} \quad (4-11a)$$

or

$$\sigma = E\epsilon \quad (4-11b)$$

The modulus of elasticity is normally referred to as Young's modulus, and materials that exhibit linear elastic behavior are described by what is called Hooke's law (equation 4-11b).

The equivalent relationship between shear stress and shearing strain is called the *shearing modulus of elasticity* or *modulus of rigidity* (G).

$$G = \frac{\tau}{\gamma} \quad (4-12)$$

Since strain is dimensionless, the units for either Young's modulus or the modulus of rigidity are the same as those used to measure the stress. Typical values of the tensile modulus of elasticity are given in table 4-1. Steel is roughly three times as "stiff" as aluminum, since E represents the slope of the stress-strain curve in the linear (elastic) region.

POISSON'S RATIO

As stated at the beginning of the discussion of strain, elongations in one direction must be accompanied by

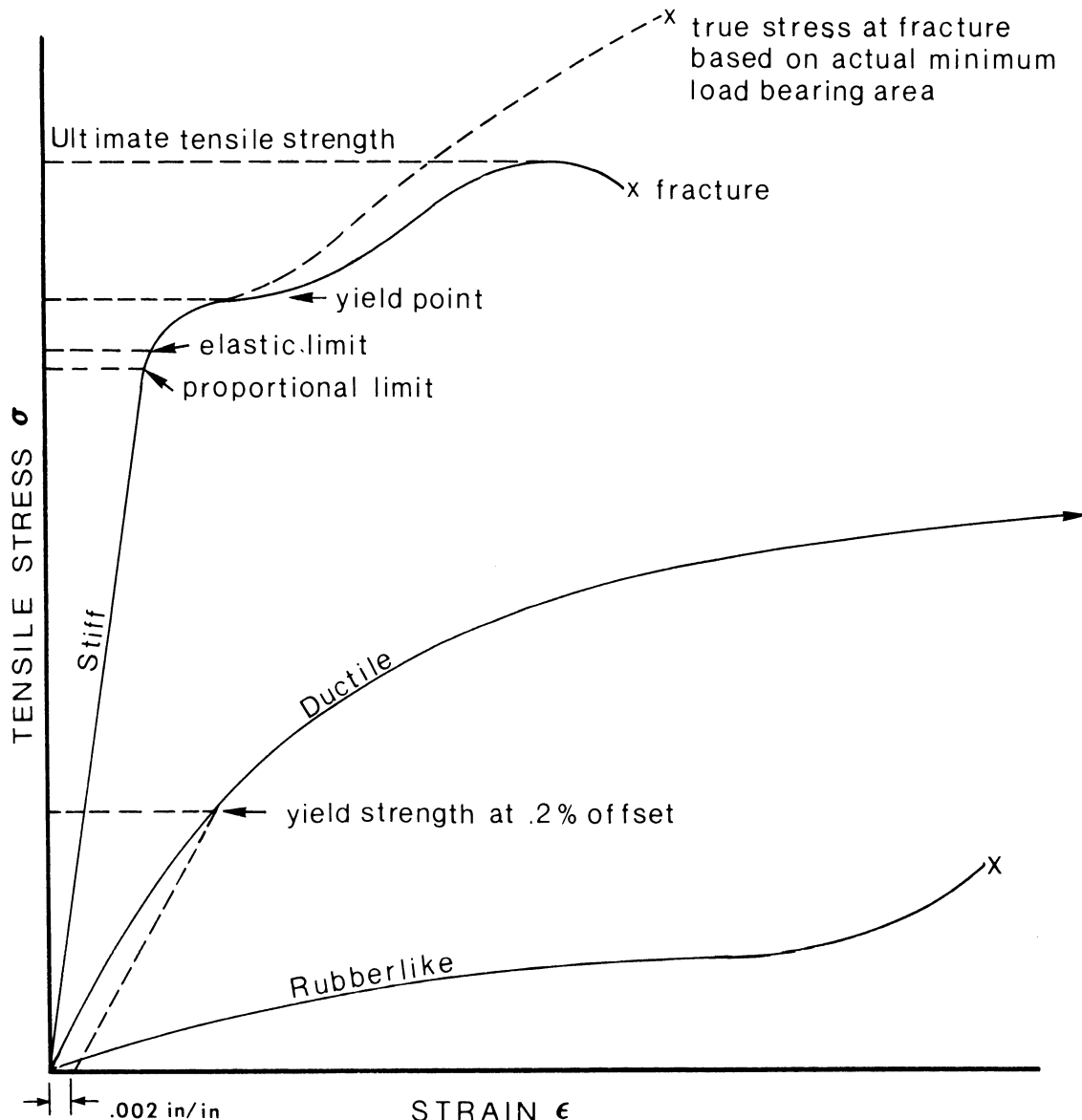


Figure 4-10. Typical stress-strain diagrams for tensile loading

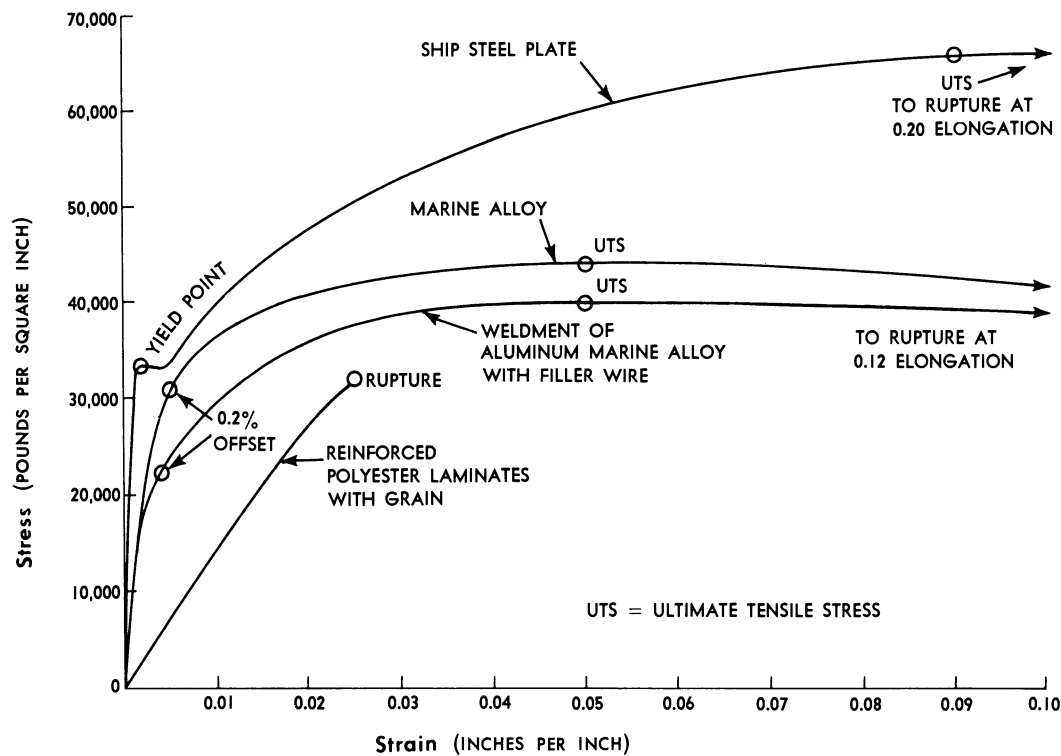


Figure 4-11. Typical stress-strain diagrams for hull materials

contractions in the other lateral directions for solid materials. If the material is loaded in one direction only, as in figures 4-5 and 4-6, the ratio of lateral (transverse) strain to axial strain will be a constant up to the proportional limit of the material. This constant of proportionality is called Poisson's ratio (μ) in honor of the French mathematician, S. D. Poisson, who discovered the constant in 1829.

$$\mu = \frac{\text{transverse strain}}{\text{axial strain}} \quad (4-13)$$

or

$$\mu = -\frac{\epsilon_y}{\epsilon_x} = -\frac{\epsilon_z}{\epsilon_x}$$

(see figures 4-5 and 4-7).

Table 4-1. Comparative Strength Characteristics of Basic Shipbuilding Materials

MATERIAL	Weight lb/cu in	TYPICAL STRENGTH (psi)			Tensile Modulus of Elasticity ($\times 10^6$)	Ratio Strength- to-Weight Col 3/Col 2 ($\times 1000$)
		Ultimate Tensile	Yield Tensile	Ultimate Shear		
(1) Aluminum alloy:						
5083-H113	0.096	46,000	33,000	27,000	10.3	480
5086-H34	0.096	47,000	37,000	27,000	10.3	490
6061-T6	0.098	45,000	40,000	30,000	10.0	460
Ship steel ASTM-A131	0.29	66,000	33,000		29.0	230
Mild steel ASTM-A100	0.28	56,000	30,000	42,000	29.0	200
Copper, hard sheet	0.32	46,000	40,000		17.0	140
(2) Reinforced polyester laminates:						
With grain	0.062	32,000		13,000	1.4	520
Across grain	0.062	21,000		14,000	1.1	340

- (1) Aluminum alloys 5083 and 5086 have excellent corrosion resistance in seawater, hence are suitable for hulls under all conditions. Alloy 6061 has good corrosion resistance in seawater, hence is suitable for framing and superstructures under seawater conditions. This alloy is also much used for the hulls of small boats for seawater use that are generally hauled from the water when not in use, or for hulls of any type that are for freshwater operation principally.
- (2) Values for polyester laminates depend upon composition and method of applying or forming. Those tabulated are typical short-term values under favorable conditions of application. Because of low ductility it is customary to design polyester laminates on basis of ultimate tensile strength after adjustment for factor of safety.

It may be shown that Poisson's ratio is related to Young's modulus and the modulus of rigidity through the equation

$$\mu = \frac{E}{2G} - 1 \quad (4-14)$$

Values of Poisson's ratio for engineering materials vary from 0.1 for concrete to 0.5 for rubber. For most metals it lies between 0.25 and 0.35.

4-3 Other Material Properties and Their Measurement

In addition to those elastic, yield, and ultimate-strength properties discussed in the previous section, other important properties are based on the plastic-deformation characteristics of various substances.

Strain Hardening

Strain hardening is an important method for increasing the yield strength of a material for design purposes. When a material specimen is loaded beyond its initial yield strength on a tensile-testing machine and then unloaded, the stress-strain curve follows a path parallel to the original elastic response down to a point on the

strain axis, which is a measure of the permanent "set" in the material (figure 4-12). When the specimen is loaded a second time, it follows the new path up to the last value of the "flow stress" on the previous test, where it then begins to yield again. The new yield stress is higher than the original yield stress, and the difference is called *strain hardening* or *work hardening*. This effect is used to advantage in bending and cold rolling of materials.

Ductility

Ductility is a measure of relative elongation, reduction in thickness, or reduction in area caused by plastically deforming a material through pulling, rolling, forging, pressing, or drawing through a die. Since strain hardening is generally involved, as the material increases in tensile strength, it decreases in ductility. One measure of ductility is to determine the "necked down" area of a specimen in a tensile test taken to fracture, as illustrated in figure 4-13.

% reduction in area

$$= \frac{(\text{original area} - \text{final minimum area})}{\text{original area}} 100$$

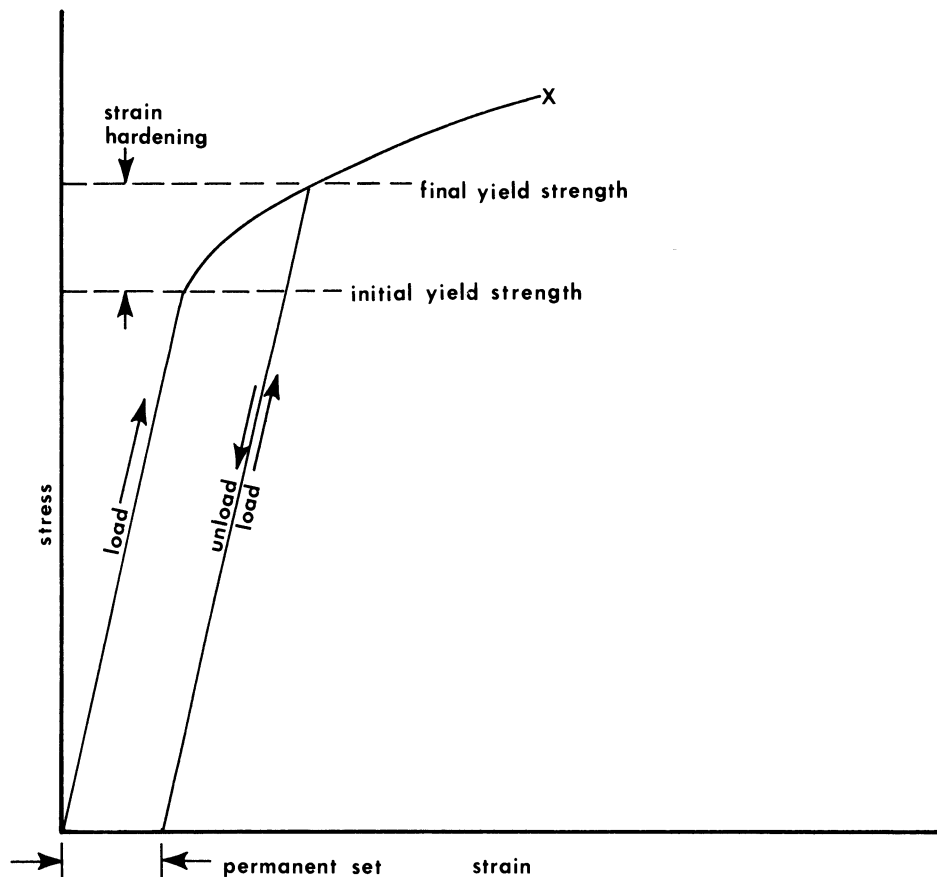


Figure 4-12. Strain hardening



Figure 4-13. Necking, or narrowing, of steel specimen at failure

Toughness

Yield strength or ultimate tensile strength do not measure the ability of a material to absorb energy before breaking. Since energy can be equated to work, which in turn is equal to force times distance, the area under a stress-strain curve is a measure of energy absorption per unit volume of the material; i.e.,

$$\begin{aligned} \text{area under stress-strain curve} &= \frac{\text{force}}{\text{area}} \times \frac{\text{distance}}{\text{distance}} \\ &= \frac{\text{force} \times \text{distance}}{\text{volume}} \end{aligned}$$

Figure 4-14 illustrates the concept of toughness or energy-absorption capacity of a material. Figure 4-14a represents the energy absorption for a brittle material that has a high tensile strength but lacks ductility. Figure 4-14b represents a ductile material that lacks high tensile strength. Figure 4-14c represents a material with good “toughness,” since it has both high tensile strength and good ductility to give the maximum area under the stress-strain curve.

MATERIAL TESTING

Tensile Tests

Most of the material properties discussed so far are measured on a tensile-testing apparatus (figure 4-15). This apparatus consists of a specimen holder, a load cell for measuring the tensile load, a moving crosshead driven hydraulically or by motor-driven screw, and an extensometer for measuring the strain. It may or may

not have automatic recording devices for taking data. A specimen of standard geometry is loaded at a pre-specified crosshead speed, and the load and corresponding elongation are recorded up to the point of rupture (figure 4-16). The data are converted to values of stress and strain and are plotted as indicated in figures 4-10 and 4-11.

Hardness Tests

The hardness test measures the resistance of a material to indentation under a specified load. The indenter is usually a ball, pyramid, or cone made of a material much harder than the material being tested. Typical indenters are made of hardened steel, tungsten carbide, or diamond for the hardest specimens. A Brinell hardness number is obtained from the indentation diameter made by a 10mm steel or tungsten carbide ball slowly pressed into the material by a Brinell hardness tester. The Rockwell hardness is indicated by the depth of the impression made by a diamond cone or hardened steel sphere applied by a standard load in a Rockwell testing machine. Hardness testing machines made by Vickers and Knoop also yield hardness numbers associated with the name of the inventor or manufacturer.

Since hardness tests are based on plastic deformation of the sample under test, manufacturing variables, such as the effect of exact carbon and alloy content and the effect of any heat treatment on yield strength, ultimate tensile strength, surface properties, and ductility, will affect both the plastic deformation properties and the hardness number. For materials that have similar work (strain) hardening characteristics, there is a good correlation between hardness and ultimate tensile strength. Thus hardness tests are very useful for checking quality control in the production of steel plate and can be used to separate mixed batches of steel plates or bars at construction sites, manufacturing plants, and shipyards.

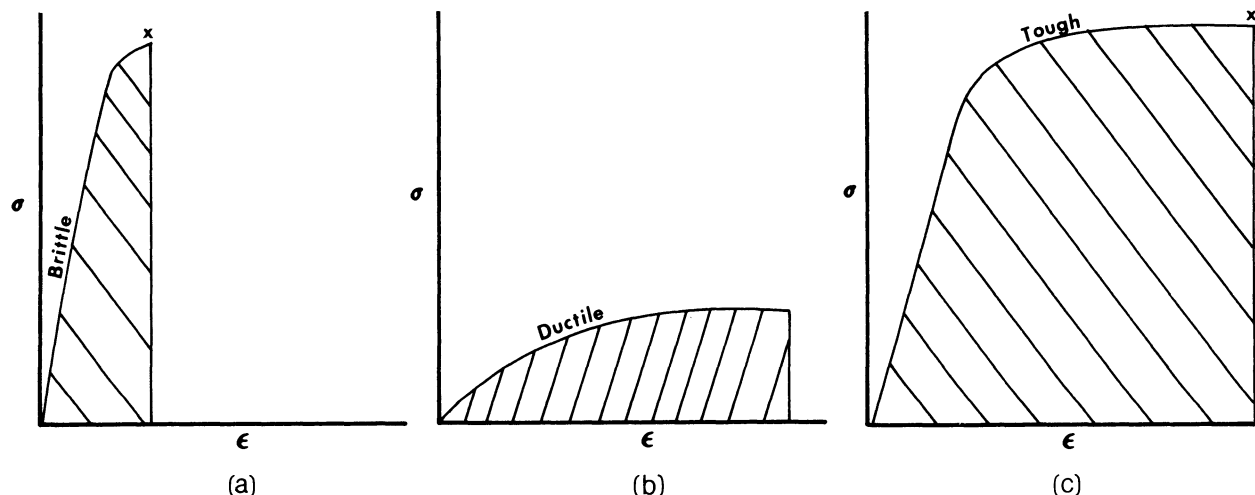


Figure 4-14. Toughness



Figure 4-15. Tensile-testing apparatus. (Courtesy of Instron Corp.)

Impact Tests

In tensile testing and hardness testing, the load is applied relatively slowly and at room temperature. Ships and marine structures, on the other hand, receive impact loads from wave action and may receive shock loads from explosions and collisions.

Some materials exhibit brittle behavior especially in areas of stress concentration when impact loads are applied, even though they appear to be ductile when loaded slowly. Other materials, such as the steels used to construct the all-welded Liberty ships during World War II, exhibit brittle behavior at low temperature. One-fourth of the nearly five thousand ships constructed of this steel developed serious cracks, especially around the corners of the rectangular cargo holds, where stress concentrations existed. In fact, 233 ships were either lost or considered unsafe as a result of brittle fracture cracks that developed at low temperature.

The repeated loads associated with heavy-weather operation undoubtedly played an important role in the loss of ships at sea. However, one of the T-2 tankers

actually broke in half at the pier on a cold night before it had seen any significant service. Failures of this sort led to an extensive investigation of the "transition temperature" range, in which the material changes its mode of fracture from ductile to brittle. By changing the chemical composition of shipbuilding steels, a higher resistance to brittle fracture at low temperature has been developed.

The method used to evaluate the notch toughness of materials over a range of temperatures is the standard notched-bar impact test. The impact test measures the energy necessary to fracture a standard notched bar by an impulse load. Figure 4-17 shows the Charpy V-notch impact tester along with the shape of the standard specimen. In the Izod impact tester, the specimen is held by an anvil and loaded like a cantilever beam. The pendulum is brought to a standard height h_0 and released. The hammer strikes the specimen at the location of the V-notch and continues to a height h' , which is a measure of the energy absorption of the material during fracture when compared to the original height h_0 . The transition temperature for brittle fracture can be de-



Figure 4-16. Tensile-test specimens showing “necked down” ductile flow before fracture

terminated by cooling the specimens to various low temperatures and quickly testing them before they warm up again. Figure 4-18 shows the impact test results for several alloys over a range of testing temperatures.

Fatigue Tests

In addition to the brittle failures caused by static, repeated, or shock loading at low temperature, many materials will eventually fail in *fatigue* at normal temperatures as a result of many alternating load cycles of peak stress below the yield strength of the material. Some ferrous materials exhibit an *endurance limit* or *fatigue limit*, defined as the stress level below which the material can endure an infinite number of loading cycles. Ductile materials such as aluminum and copper do not exhibit a *true* endurance limit and will eventually fail if the combination of peak stress and number of cycles is sufficiently large. For these materials, the stress corresponding to some arbitrary number of cycles is taken as the endurance limit.

In general, the smaller the peak stress, the greater the number of cycles required for the material to fail in fatigue (figure 4-19).

Material specimens can be tested on a wide variety of fatigue-testing machines, and some structures such as airplane wings are often tested at full-scale to determine the *low-cycle fatigue failure* characteristics of a

complete structure with many parts subject to fatigue loading. The U.S. Navy has conducted controlled fatigue tests on small full-scale aluminum-hulled ships, but it is not possible to conduct such tests on large ships. Fatigue testing is therefore limited to checking out sections of the hull that are subject to stress concentrations and therefore likely candidates for fatigue failure. Candidate materials for the cargo tanks, such as those used to carry liquified natural gas (LNG) and other cold substances, must be carefully tested for fatigue and brittle fracture at low temperatures before being used for the construction of such tanks.

4-4 Ship Hull Materials

The age of “iron men in wooden ships” lasted many centuries. The age of iron-clad ships and iron ships was very brief, lasting from the 1830s until steel-production techniques were developed in the 1890s. Although wood is still used for minesweepers and commercial and pleasure craft, cast and wrought iron has virtually disappeared as a material for hull construction. Brittleness and difficulty of fabrication have made iron obsolete as a ship hull material.

Modern steels are characterized by high strength, good ductility, and ease of fabrication. Steel has dominated the ship-construction business since it was introduced, although aluminum, titanium, reinforced con-

crete, and fiberglass-reinforced plastics all have their advantages for selected applications. The type of material selected is usually dictated by the relative priorities assigned to cost, strength, density, corrosion resistance, brittle-fracture resistance, and other properties discussed in this chapter.

STEEL IN SHIPS

Since the development of the Bessemer and, more importantly, the open-hearth processes of manufacture, the principal material for ship hulls has been steel in its various forms and alloys.

Steel is a homogeneous material with excellent strength characteristics; it may be cast, forged, or hot or cold rolled. It can be fabricated into complex structures using various welding processes, and the weldments are uniform and reliable.

The primary disadvantage of steel for shipbuilding use is its lack of resistance to corrosion. It is especially vulnerable to corrosion in the presence of seawater, and therefore requires careful attention when being painted. It also requires constant vigilance in hull maintenance. Cathodic protection is often installed in the vicinity of

the propeller(s) and rudder as an adjunct to hull coatings.

Low carbon steel (.16 to .23 percent carbon) is the standard material for economical ship-hull construction. These ordinary strength (mild) steels are specified by American Bureau of Shipping (ABS) standards as to process of manufacture, deoxidation method, chemical composition, heat treatment, tensile-test properties, and impact-test properties. (U.S. Navy specifications essentially correspond to ABS standards.) The minimum yield point is 34,000 psi, and the tensile strength is between 58,000 and 71,000 psi.

High-tensile steel (HTS) is used in many naval applications where structural members may be subjected to larger than normal stresses. Additional alloys such as manganese are added to the low carbon content, and the steel is normalized by heat treatment to raise its yield strength to around 50,000 psi and the UTS to 70,000–90,000 psi. This steel was used extensively in submarine construction until the development of high-yield steel (HY). HTS was used in the construction of the pressure hull of the first nuclear-powered submarine, the U.S.S. *Nautilus*. It is used on surface ships in high-stress areas to reduce structural weight.

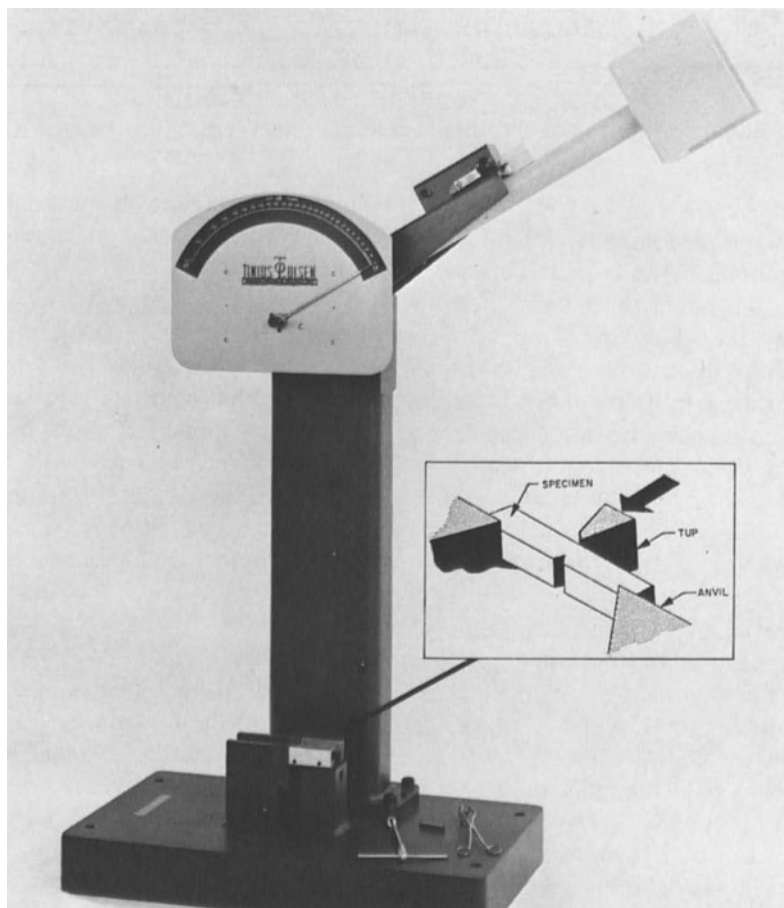


Figure 4-17. Standard impact-testing apparatus. (Courtesy of Tinius Olsen Corp.)

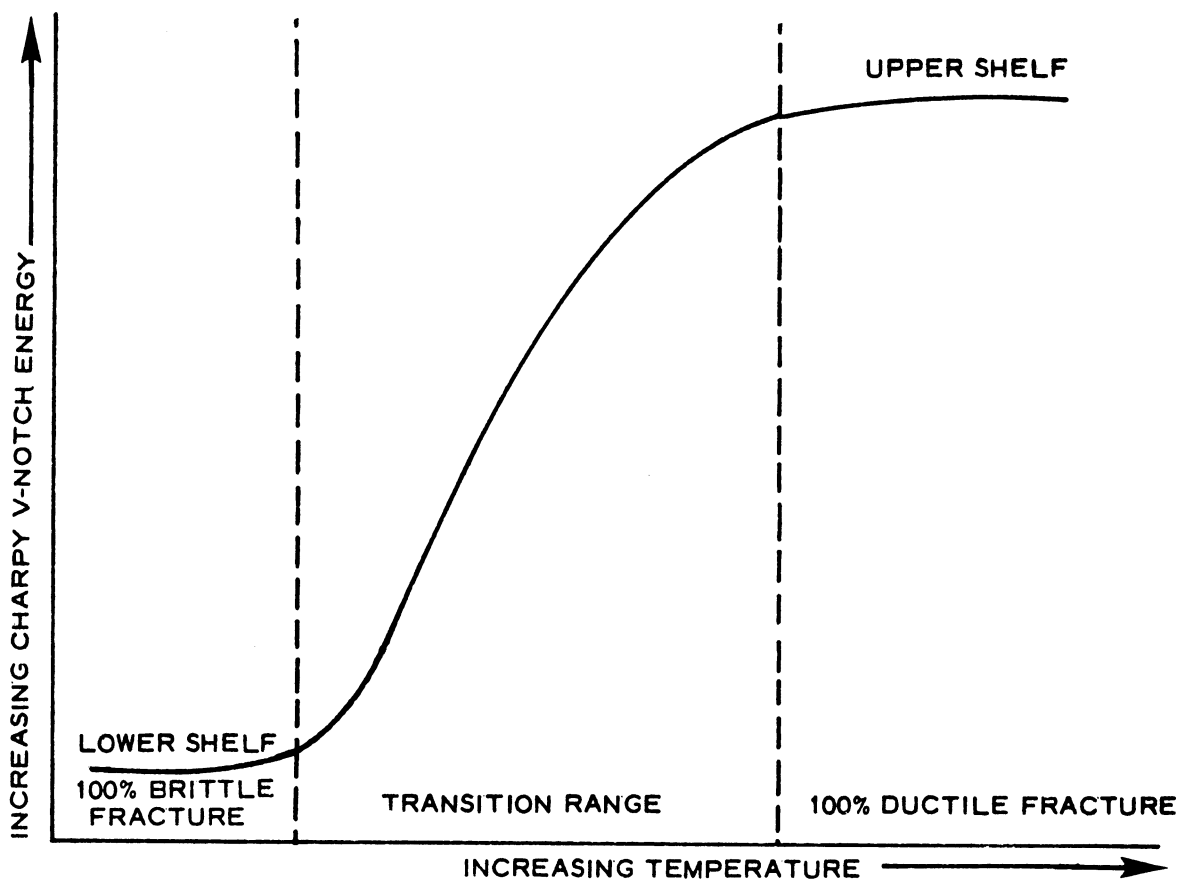


Figure 4-18. Variation of notch toughness with temperature. (From Taggart 1980.)

High-yield quenched and tempered steel (HY steel) has been developed for several naval applications. It is the principal steel for submarine-hull construction. HY-80 was the first low-carbon steel to have its strength and toughness increased by quenching and tempering. This rapid cooling of the hot steel to control the microstructure, followed by reheating (annealing) to relieve the residual stresses, makes the steel more ductile. This process also increases its impact strength (notch toughness). The development of HY-80 with a yield strength of 80,000 psi was followed by the development of HY-100 and HY-130, with yield strengths of 100,000 psi and 130,000 psi respectively. Since the strength of these steels depends on the heat treatment during production, they are more complicated to weld while maintaining their strength. HY-130 was used in the construction of the pressure hull for the U.S. Navy's Deep Submergence Rescue Vehicle (DSRV).

Ultra high-strength steels of yield strengths to 100,000 psi and ultimate tensile strengths of 135,000 psi are used in the critical locations of the hull structure, such as deck stringers and sheer-strake plating (see figure 5-32). Steels less high-strength but still greater strength than mild steel are frequently used in the bilge strake,

where their toughness properties eliminate the need for the conventional riveted crack-arrestor strake in this region.

The stability of ships built with variations in strength properties of steels is considerably improved by an overall weight reduction of 17 percent and a 50-percent weight reduction in topside steel. Reduction in propulsion machinery for lighter, high-strength steel vessels saves on initial costs and still allows the speed-power requirements to be met. With the correspondingly smaller power plants, which have lower fuel consumption, operating costs are reduced accordingly.

Special Steels

Special steels are required when the operating environment justifies the high cost and difficulty of fabrication that most of them exhibit.

Corrosion-resistant steels (CRS, or more commonly, stainless steels) have been developed for situations where corrosion must be avoided, such as cargo tanks carrying corrosive chemicals or potable liquids like fresh water or wine. CRS steels contain chromium and nickel and have reduced carbon content (0.03–0.08 percent). Some stainless steels are difficult to machine and difficult to

weld, so the installation costs as well as the material costs are high.

Steels for low-temperature application are required for refrigerated cargo ships, liquified natural gas carriers (LNG ships), and icebreakers and mobile offshore drilling rigs located in very cold regions. Special attention must be placed on notch toughness at very low temperature to avoid the problems of brittle fracture.

For ballistic decks and where protective bulkheads and shields (armor plating) are required, *special-treatment steel* (STS) is employed. This is a nickel-steel alloy that has the excellent strength characteristics required for armor plate. The presence of nickel increases hardness, toughness, and elasticity while it decreases ductility only slightly.

Manufacturing

Steel is manufactured in various forms and, in most cases for structural members and hull plating, may be obtained from mills in the forms desired. Where special shapes or sizes are required (in naval construction the requirements are numerous) arrangements are made with the steel mill for supply if possible; otherwise, they are made from the standard sizes or fabricated by shipyard facilities. Common examples of this are the cutting of I-beams to T-bars, or channels to angles.

Steel comes from the mill generally in two basic forms: plates and shapes. Plates are steel flat stock of a quarter inch or more in thickness (steel less than a quarter of an inch is sheet steel). They vary in width from approximately 60 inches to 110 inches. Plate is normally designated by its weight in lbs/ft² and varies normally from about 10 to over 40 pounds plate, although the

plate used in the pressure hull of a submarine is considerably heavier. The density of steel is approximately 490 lbs/ft³; a plate one inch in thickness would be 40.8 pound plate, one-quarter inch, 10.2 pound plate, etc. (Normally the decimals are omitted in referring to the plate sizes.) Below 10 pound plate, sizes are given in gauges.

Shapes are normally supplied as flat bars, plain-angle bars, beveled-angle bars, bulb plate, T-bars, T-bulb bars, channels, and I- or H-beams.

NONFERROUS METALS IN SHIPS

With the exception of aluminum and its alloys as noted here, practically no structural use is made of any other nonferrous metal in shipbuilding. However, titanium is being increasingly used in naval craft where extreme strength and light weight are urgent requirements (see below).

Aluminum

An excellent selection of aluminum alloys is available for shipbuilding use with a range of characteristic properties that are more adaptable than steel in many ways. The primary advantage and special characteristics of aluminum are its light weight and its superiority under bending stresses for special structures applications. In addition, it is superior to other metals in its greater ductility, distortion strengthening, and impact resistance. It is nonsparking and nonmagnetic. In certain of its alloys aluminum is highly resistant to corrosion. Unfortunately, it is more expensive than steel and more expensive to work. Welding techniques for aluminum have continuously been improved, but it is still more difficult to weld than ordinary steel.

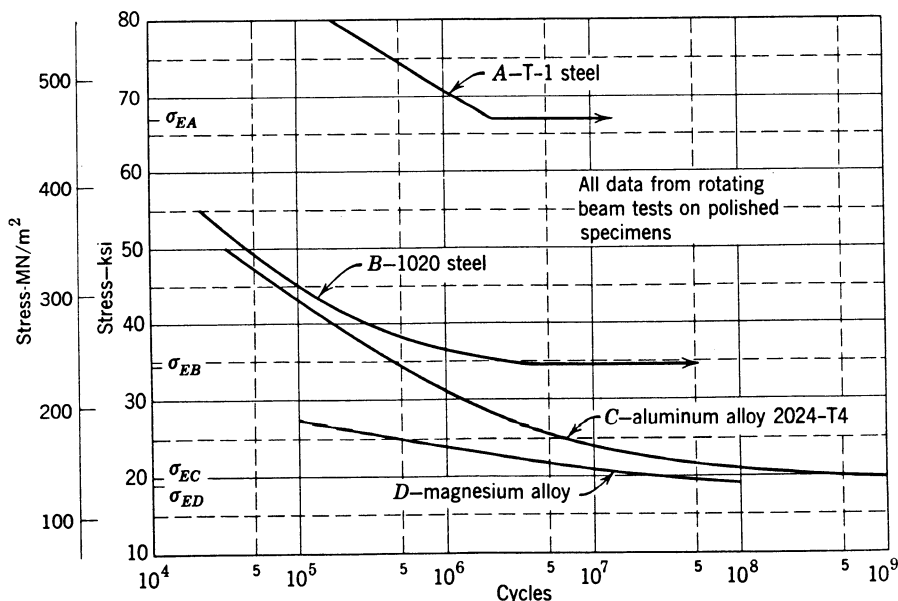


Figure 4-19. Fatigue endurance limits. (From Higdon et. al. 1976.)

Aluminum has a relatively low endurance strength, about one-third of its tensile-strength value. The effective fatigue limit is reached at the end of about 5×10^8 cycles. Because of this and because of the low modulus of elasticity of aluminum (10×10^6 psi), vibration control of aluminum structures is more important than that of steel structures. Aluminum, when used together with copper-based alloys in salt water, is also subjected to a high rate of galvanic corrosion. In such cases special corrosion-control methods, such as the use of zinc sacrificial blocks, are required.

While aluminum has been used in the superstructures of many large vessels, such as passenger liners and destroyer-type ships, its use for the entire hull structure has, to date, been confined to relatively small vessels where weight is critical. Such vessels are hydrofoil-supported craft, high-speed coastal gunboats, torpedo boats, surface-effect craft, yachts, etc.

Aluminum would probably be in greater use for small craft had it not been surpassed in economy of fabrication by the ever-expanding use of reinforced polyester laminates. The relative strength properties of these primary shipbuilding materials are given in table 4-1.

Other Materials

Titanium, fiberglass, wood, and concrete are all used as ship-hull materials. Titanium is expensive, but it has a high strength (yield stress of 150,000 psi) and low specific gravity (4.5), which make it very attractive for weight-sensitive applications such as the pressure hulls of deep submergence vehicles. Titanium also has very high corrosion resistance. For this reason it is used where surface coating for corrosion control is not possible, such as for propellers and valves. The welding of titanium is expensive and requires a carefully controlled procedure. The fatigue strength of titanium is about half of its yield strength, and its notch sensitivity is high compared to that of aluminum.

Composites such as fiberglass-reinforced plastic offer exceptional corrosion resistance and high strength-to-weight ratio. The fatigue strength is about 25 percent of its maximum yield strength. High temperatures and ultraviolet light also decrease the mechanical properties of fiberglass.

Ferro cement and concrete are sometimes used in the construction of small vessels, barges, and other ocean structures. Concrete has good strength in compression (about 7,000 psi) but low strength in tension. Adequate reinforcement by steel wire mesh, commonly called ferro cement, is used to bring the combined tensile strength of the structure to an adequate level. Concrete resists high temperatures and corrosive environments well, but deteriorates as absorbed water freezes in it.

Earlier, wood was the only hull material available, having an average yield stress of 2,500 psi. Because of

cost, availability, and preservation problems it is finding only limited application today, mainly in the construction of special vessels. Wood has been used in shipbuilding for more than 5,000 years. Wooden seagoing vessels will probably be in use as long as remote shipyards and primitive communities continue to exist.

There is presently a resurgence in the construction of wooden vessels, a result of renewed appreciation of old shipbuilding skills. More wooden yachts were built in 1980 than in the whole decade preceding. It is worth noting that there are qualities in well-constructed wooden vessels not found in ships of any other material. Such things as thermal insulation and acoustic insulation make a wooden ship a more comfortable habitat than the customary modern ship environment, which is plagued by the noises of engines and water and the extremities of temperatures at sea.

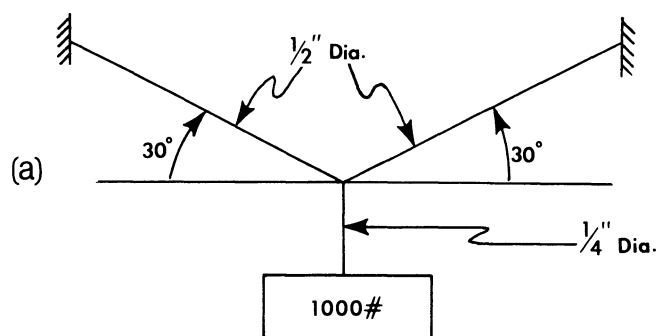
New methods and new adhesives have also resulted in the production of laminated wood in a cold molded process. Very successful lightweight ocean-racing yacht hulls have been built of such material. The saturation of wood fibre with the adhesive brings the material properties of the final composite material very close to those of other reinforced plastics, such as fiberglass and kevlar. The advantage of such cold molded wood is in its low-weight strength.

In considering all the types of materials in use for structural applications in ships, the focus is still essentially on steel for vessels of all sizes. For large vessels, because of steel's greater strength and relative economy, there is at present no other choice.

An excellent discussion of ship-hull materials and welding is found in *Ship Design and Construction* (chapter 8).

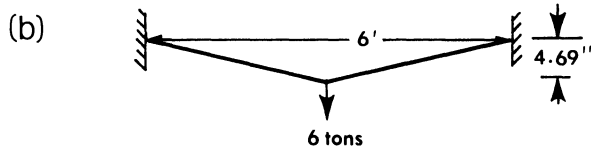
Problems

4-1. Determine the stress in the 0.50-in- and 0.25-in-diameter cables shown in figure 4-a. If the mod-



ulus of elasticity E equals 10×10^6 psi for the 0.50-in-diameter cable and its total length before application of the 1,000-lb load was 8 feet, what is the elongation of the 0.50-in cable after application of the 1,000-lb load?

4-2. A length of #16 gauge (0.51-in diameter) music wire is stretched between supports that are 6.0 feet apart, as illustrated in figure 4-b. When a weight of 6.0



tons is suspended from the wire, its center deflects 4.69 inches. The proportional limit for the wire is 266,000 psi. Neglect any initial horizontal load in the wire and find

- the stress in the wire
- the total elongation of the wire under the 6-ton load
- the Young's modulus of the wire

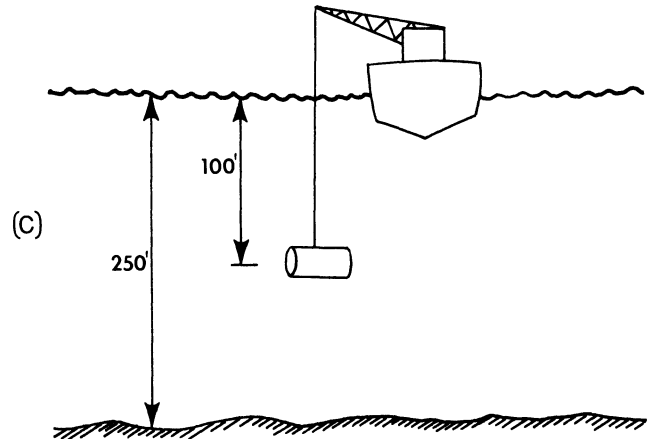
4-3. A 0.50-in-diameter rod starts to plastically deform when the load it is supporting exceeds 15,750 pounds. What is the yield strength of the metal specimen? Determine the elongation of a 2-in-long section of the specimen if the modulus of elasticity is 30×10^6 psi.

4-4. A tensile test specimen is loaded to 25,000 psi. The 2-in gauge length elastically increases in length by 0.0017 inches. Determine the modulus of elasticity of the specimen.

4-5. A 0.0625-in-diameter wire 10 feet long increases in length elastically by 0.30 inch when it supports a 75-lb load. Determine the Young's modulus of this material.

4-6. A cylindrical undersea habitat is being lifted from the ocean bottom by a 0.50-in-diameter steel cable that is attached to a shipboard crane. The habitat is lifted to a position 100 feet below the surface when the

lifting operation is halted momentarily. The habitat has a diameter of 6 feet and a length of 14 feet (outside dimensions). The habitat weighs 33,000 pounds and the total depth of water is 250 feet. The modulus of elasticity for steel is equal to 30×10^6 psi.



- If the habitat is in equilibrium at the position shown in figure 4-c, what is the buoyant force acting on it?
- What is the strain in the steel cable (assume no plastic deformation)?
- When the habitat is lifted completely out of the water (but still suspended from the steel cable), what type of behavior will the cable exhibit if $\sigma_y = 60,000$ psi and $\sigma_{ult} = 82,000$ psi?
 - elastic deformation
 - plastic deformation
 - proportional deformation
 - fracture

The Strength and Structure of Ships

5-1 Strength

The structure of ships is more complex than most other man-made structures for several reasons. In addition to satisfying the general arrangements required by its mission and payload (and tonnage measurement rules for merchant ships), the exterior hull and internal tankage must be watertight structural envelopes able to withstand the *anticipated loads* encountered at sea. The exterior hull must also conform to the rules of good hydrodynamic design practice, which involves more complex geometries than are customarily found in land-borne structures.

It is in the area of anticipated loads that the most difficulty is encountered. These loads are the static and dynamic loads of the weights of cargo, machinery, and structures as well as the buoyant force of the seawater and the environmental loads of wind, waves, ice, and thermal effects. The dynamic environmental loads, especially the wave loadings in a hostile sea, are *indeterminate* in nature and can only be described in a statistical sense. These external and internal loads, resulting from the relative motion between the ship and the sea, require that the ship structure withstand the impact loads of slamming and deck wetness, the wave-induced vibrations of propellers and machinery, fatigue from repeated bending, and many other phenomena associated with seagoing vessels.

At present, the approach to structural design is considerably enhanced by extended research into the statistical description and effects of sea conditions, together with the powerful analytical capabilities of digital computers. With this combination of research and computers, an attempt is being made to answer complex questions concerning the optimization of structural design to satisfy both strength and cost requirements.

For this introduction to the design of ship structures, ship strength will be approached by the traditional route,

with methods that have long been used and are still the basis of structural design. More recent applications of strength studies will then be considered as extensions of the basic approach described.

A ship in a seaway can be considered similar to a beam with supports and distributed loads. The supports are buoyant forces of the waves and the loads are those of the weight of the ship's structure and load within, such as fuel, water, and cargo. The worst condition of loading and support for a ship occurs when it heads into or away from the sea, with waves approximately as long as the length of the ship. A quartering sea can also produce this condition if the ship's bow and stern are either in troughs or on crests at the same time, in which case torsional loads must also be considered.

The ship shown in figure 5-1 is supported by waves, with the bow and stern each riding a crest and the midship region in the trough. This ship will bend with compression at the top and tension at the bottom. The ship is said to be *sagging*, and in this condition the weather deck wants to buckle due to compressive stress, while the bottom plating stretches due to tensile stress.

When the ship advances half a wave length, so that the crest is midships and the bow and stern are over troughs, as in figure 5-2, the stresses are reversed. The weather deck is in tension and the bottom plating is in compression, and the ship is said to be *hogging* (as in carrying a pig over your shoulder).

SHIP STRENGTH AND THE BEAM THEORY

The beam theory assumes that a ship, for purposes of strength considerations, may be likened to a hollow, nearly rectangular girder. Basically, the comparison is correct. Assuming first continuity of the material structure and second the known distribution of forces, calculations can be made for the strength of such a beam of known cross section, regardless of size. However,

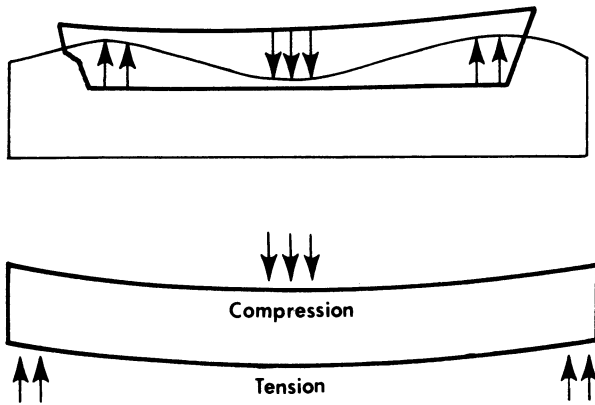


Figure 5-1. Sagging condition

because of the complexity of the ship's structure, the discontinuity of various members, the variations in fastenings (rivets and welding), the openings in the hull, the dynamic loads as well as complex static loads imposed, to cite but a few of the discrepancies, the simple-beam theory becomes a general rather than an exact criterion. This is not to say that it is of little value—beam theory is a primary tool used in ship design. In all cases of structural damage, beam theory should be the basic method employed in analyzing any reduction in structural strength and the basic guide in considering restorative measures.

Although discrepancies exist in assuming the ship is a simple girder, beam theory has produced a reliable basis upon which an analysis may be made. It is an approach universally used, and hence provides a standard upon which strength calculations may be made, and upon which results may be analyzed and compared.

5-2 Beam and Load Classification

Before discussing the application of beam theory to ship problems, the essential elements of the study of solid mechanics (strength of materials) will be presented. This involves analysis of the effects of different types of loads

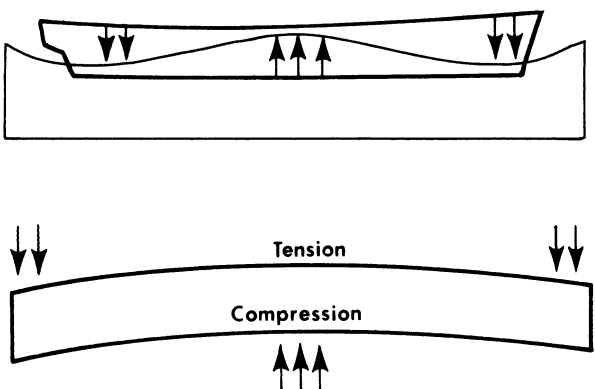


Figure 5-2. Hogging condition

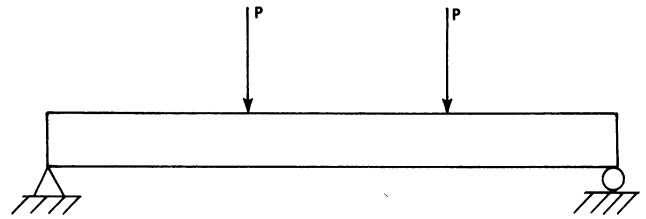


Figure 5-3a. Concentrated loads on a simply supported beam

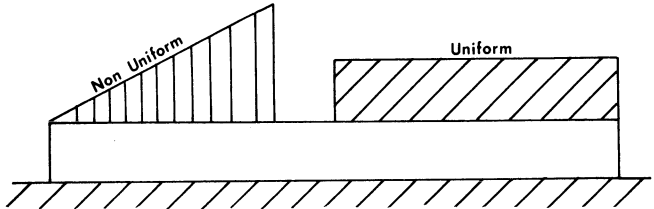


Figure 5-3b. Distributed loads on a continuously supported beam

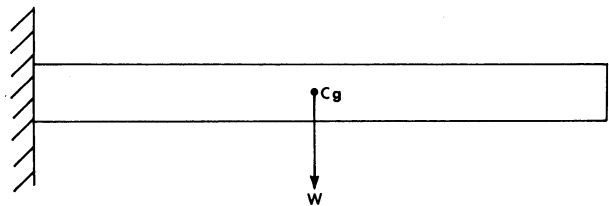


Figure 5-3c. Gravitational load on a cantilevered beam

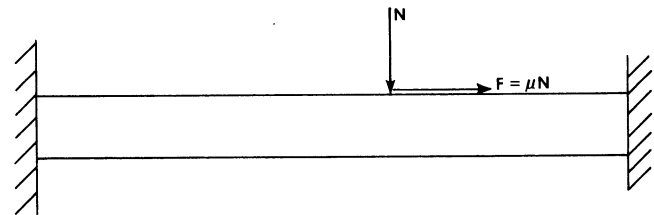


Figure 5-3d. Frictional and concentrated loads on a fixed end beam

on beams supported in various standard configurations, as illustrated in figure 5-3. (The support classifications were covered in section 2-5 of chapter 2.)

1. *Concentrated loads* or point loads are considered to exist when the area of contact is small relative to the size of the beam.

2. *Distributed loads* are considered to exist when the area of contact is large relative to the size of the beam. Distributed loads may be uniform and stated in terms of the load per unit length of the beam. They may also vary uniformly according to some mathematical relationship or they may vary in an arbitrary fashion.

Special cases of the above loads include

3. *Frictional loads*, which act parallel to the area of contact and are generally a function of the normal force times the coefficient of friction.

4. *Gravity loads*, which are those caused by the weight of the beam and may either be considered as a distributed load or a concentrated load at the center of gravity

of the beam. To simplify some problems, the beam is considered to be weightless (as in example 2-2).

5. *Thermal, inertial, and magnetic loads*, which, like gravity loads, do not depend on body contact, but apply throughout the beam.

Loads may also be classified with respect to the types of deformation they produce or with respect to the time of action of the load.

1. *Axial loads* pass through the centroid of the beam sections and produce tensile or compressive strains, as discussed in chapter 4.

2. *Torsional loads* or torque loads cause the loaded member to twist or rotate relative to itself about some axis. Torsional loads are found in shafts used to transmit power and in the suspensions of some automobiles and racing cars.

3. *Bending or flexural loads* are caused by forces that produce moments and couples on a beam and result in variations of stress and strain across a section of the beam.

4. *Shearing loads* are those that produce shearing strains in a beam.

5. *Combined loads* are combinations of the above that produce complex deformations of the structural member.

In terms of the time of action of the load, the classifications are

1. *Static loads*, which are applied gradually and may or may not be sustained over a long period of time (*dead loads*). Static equilibrium is maintained unless failure occurs.

2. *Dynamic loads*, which vary with time and generally fit the following categories, discussed in chapter 4:

- a) Repeated or fatigue loads, which involve large numbers of cyclic loading over a period of time. Resonant vibrations in the structure may or may not be induced, depending on the frequency of the repeated load.
- b) Impact or energy loads, which are rapidly applied loads producing vibrations and occasionally permanent deformation in the structure. Equilibrium is not reestablished until the vibrations dampen out.

5-3 Load and Reaction Analysis in Beams

The technique for the analysis of concentrated loads and their reactions was developed in section 2-5 of chapter 2 on static equilibrium. The analysis of distributed loads caused by hydrostatic forces was covered in section 2-8 of chapter 2. The same technique of using force and moment summations can be applied to beam problems if the distributed loads are considered to act at the centroid of the area that represents the distribution of the load.

Example 5-1: Determine the support reactions for a simple beam loaded as illustrated in figure 5-4. Assume the beam is weightless. The applied loads are given in *kips*, which are equivalent to 1,000-lb increments (kilopounds).

Solution: The distributed loads can be assumed to be equivalent to a concentrated load equal to the area under the cross-hatched representation of the distributed load and acting at the centroid of the area. The uniformly distributed load on the left-hand side is therefore equal to 20 kips acting 5 feet to the left of rolling support *A*, and the uniformly varying load on the right is equivalent to a concentrated load of 12 kips acting at the centroid of the triangle, which is 4 feet to the left of pinned support *B*.

A free-body diagram is drawn in figure 5-4b, showing the angled force of 5 kips replaced by its horizontal and vertical components and the reaction at *B* replaced by the horizontal and vertical components. The rolling support at *A* has only a vertical reaction force.

Since there are only three unknown reaction forces, the problem is *statically determinate*: There are three independent equations to be solved for the three unknowns.

$$\sum F_x = 0$$

$$3k - R_{Bx} = 0$$

$$R_{Bx} = \boxed{3 \text{ kips } \uparrow}$$

$$\sum M_A = 0$$

$$R_{By}(15) + 20(5) - 4(5) - 12(11) = 0$$

$$R_{By} = \frac{132 + 20 - 100}{15} = \boxed{3.47 \text{ kips } \uparrow}$$

$$\sum M_B = 0$$

$$20(20) + 4(10) + 12(4) - R_{Ay}(15) = 0$$

$$R_{Ay} = \frac{400 + 40 + 48}{15} = \boxed{32.53 \text{ kips } \uparrow}$$

Check:

$$\sum F_y = 0$$

$$R_{Ay} + R_{By} - 20 - 4 - 12 = 0$$

$$32.53 + 3.47 - 36 = 0$$

$$0 = 0$$

5-4 Load, Shear, and Bending-Moment Diagrams

Before the local stress distribution, including the maximum stress, can be calculated in a loaded beam, the variation in bending moment along the beam must be

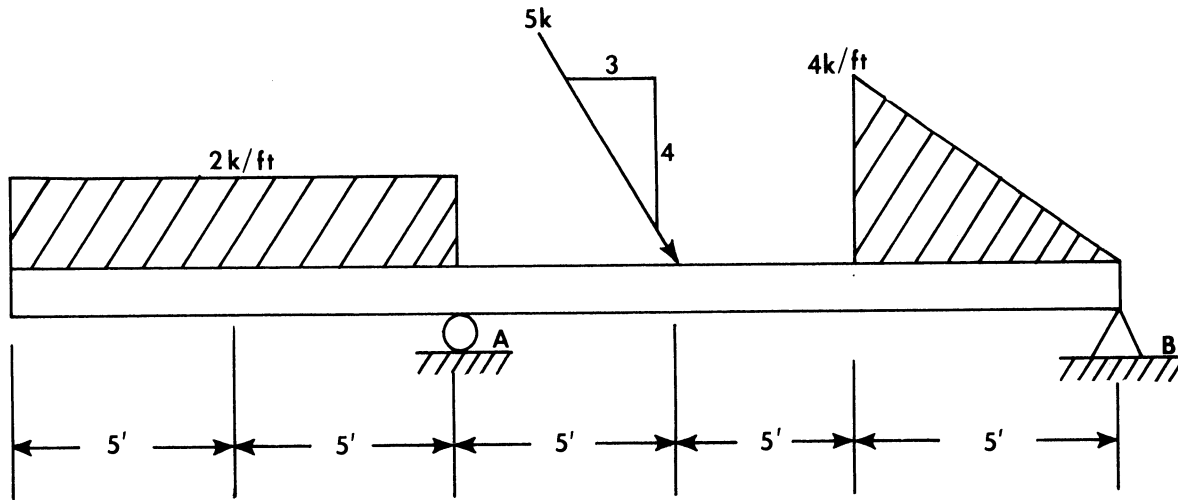


Figure 5-4a. Beam loads for example 5-1

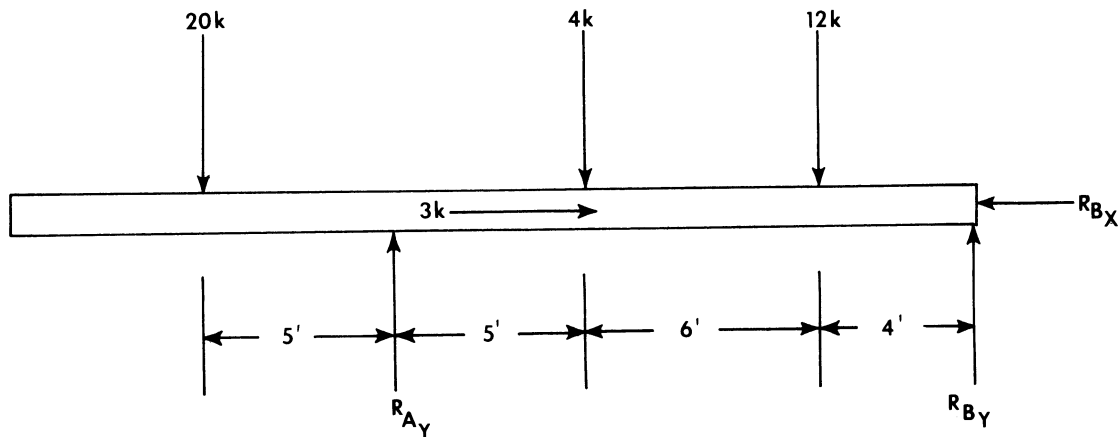


Figure 5-4b. Free-body diagram of example 5-1

analyzed. This can be accomplished by sectioning the beam at various locations and summing moments to the right or left of the section, as illustrated in figure 5-5. Since static equilibrium is maintained in a stationary beam, an *internal shear force* and an *internal resisting moment* that counteract the external forces and external bending moments must act at the face of the section. The magnitude of the resisting shearing force V (sometimes denoted by F_s or S) is found by summing forces on the free-body diagram of the sectioned beam in the vertical direction. The magnitude of bending moment at a section is found by summing moments about the section. In the example of figure 5-5a, the internal shear force is of constant magnitude and opposite direction on either side of the concentrated force $2P$ and may be plotted as in figure 5-5b. The bending moment, on the other hand, varies linearly with distance from either end reaction, as illustrated in figure 5-5b.

Rather than the calculation of the value of shear and bending moment for each section in the beam, a graphical technique is frequently used, based on the relationship between the load, shear, and bending moment, suggested by figure 5-5b. Since the bending moment at any point is the algebraic sum of the moments of the forces to the left of the section, bending moment is equal to the area under the shear curve up to a particular section; that is,

$$M(x) = \int V dx + c \quad (5-1)$$

or

$$V(x) = \frac{dM}{dx} \quad (5-2)$$

Thus the value of the shear force represents the slope of the bending-moment curve. The sign convention for

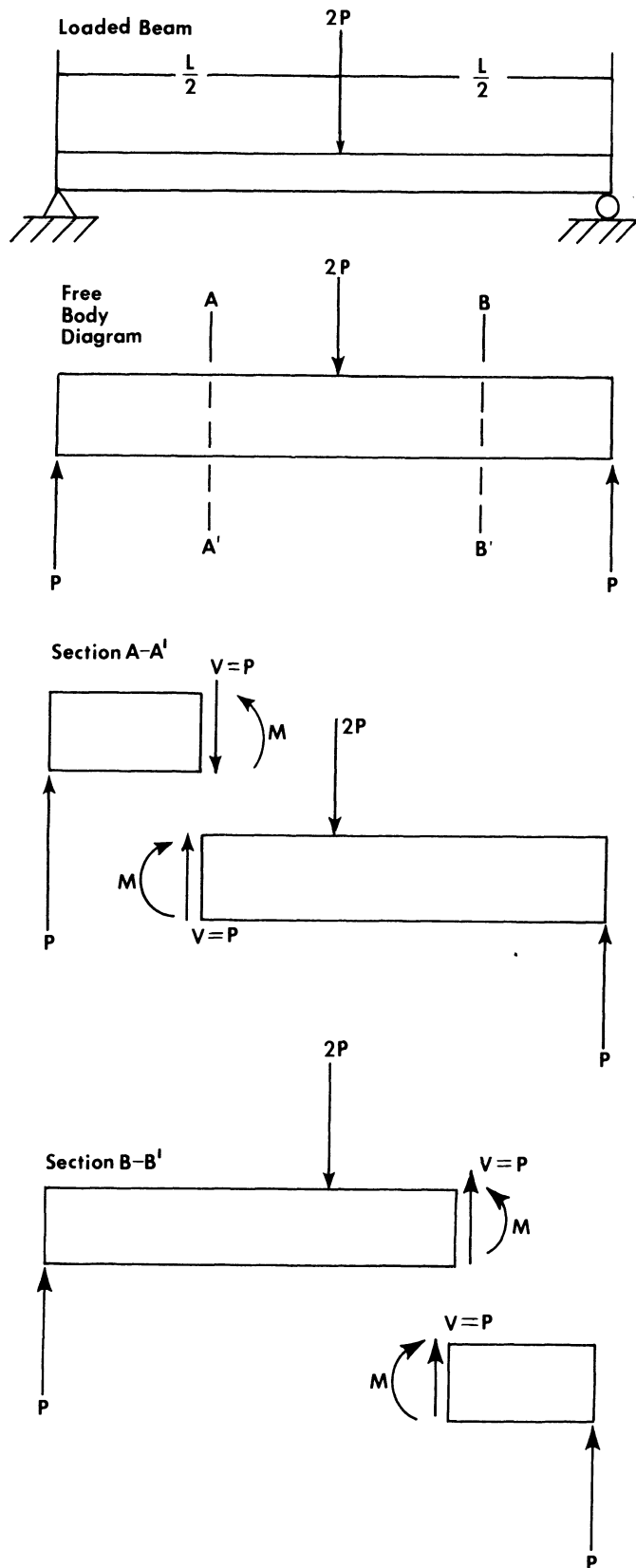


Figure 5-5a. Simple beam with a concentrated load

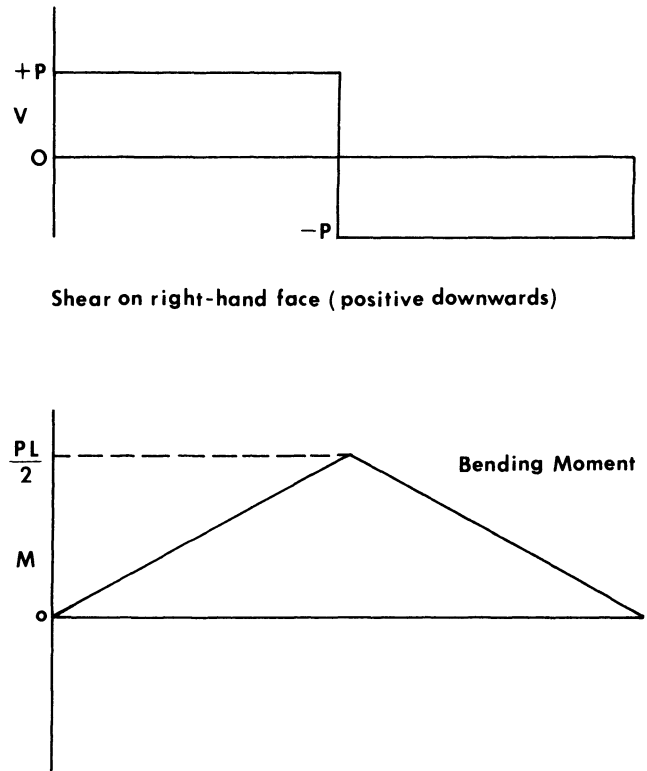


Figure 5-5b. Shear and moment diagrams

equation 5-2 is arbitrary and corresponds to the sketch in figure 5-6.

It can be shown that the shear force is the algebraic sum of the vertical loads on the free body to the left of the section and that the slope of the shear curve is given by the value of the distributed load per unit length at that section.

$$V(x) = \int w dx + C \quad (5-3)$$

or

$$w(x) = \frac{dV}{dx} \quad (5-4)$$

where $w(x)$ = the load per unit length taken positive upwards.

Concentrated loads appear as vertical lines on the shear diagram because the slope becomes infinite as the contact area approaches zero. The construction of the load, shear, and moment for a combination of concentrated and distributed loads is illustrated in figure 5-7 for example 5-2.

Example 5-2: A 20-ft-long weightless beam supported at two points 10 feet apart is uniformly loaded at 10 tons per foot along the entire length of the beam. Calculate the reactions and draw the load, shear, and moment diagrams.

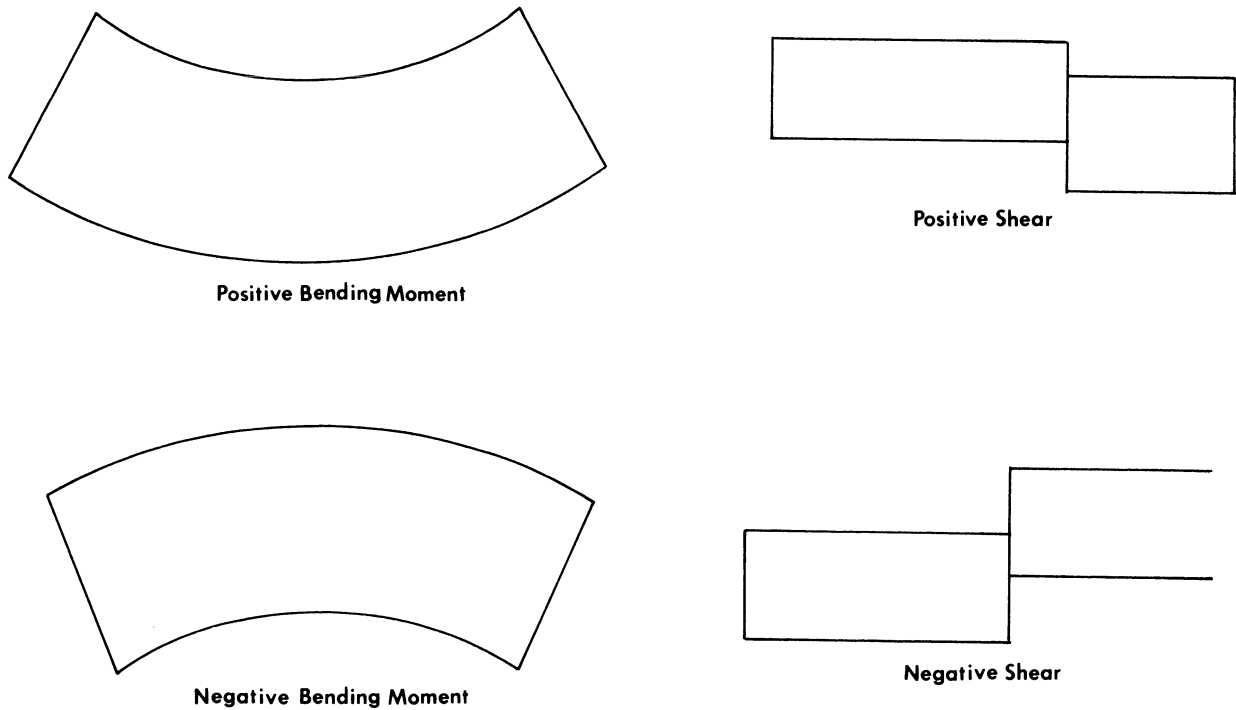


Figure 5-6. Sign convention

Solution: The reaction forces are equal since the beam is symmetrically loaded with a total of 200 tons. This is divided into 100-ton reactions at *A* and *B*. The load, shear, and bending-moment diagrams are then drawn by summing the areas under the curve at all significant points in the shear curve. For example, the value of the shear at *x* equals 5' is

$$V = \int w dx = (-10 \text{ tons/ft})(5 \text{ ft}) = \boxed{-50 \text{ tons}}$$

The value of the bending moment at *x* equals 5' is

$$M = \int V dx = -50 \text{ tons} (5 \text{ ft})/2 = \boxed{-125 \text{ ft-tons}}$$

The concentrated load at *x* equals 5' raises the shear to +50 tons, at which point it starts to fall again at 10 tons/ft. Note the break in curvature on the bending-moment curve at the sudden change in slope of the shear curve. Also note that the bending-moment curve has a maximum or minimum each time the shear curve crosses zero.

For problems involving no external couples and no end-of-beam reactions, the areas above and below the shear curve balance out, resulting in zero moment at the ends of the beam. If one of the beam ends is fixed, the summed moment calculation must equal the moment calculated in evaluating the support reactions. These "boundary conditions" provide a check on the graphical solution.

5-5 Longitudinal Bending Moments and Strength Curves for Ships

If a body of constant cross section made up of homogeneous particles is floating in still water, there will be no bending moment because the body is in static equilibrium, and the uniformly distributed weight is balanced for each increment of length by the uniformly distributed buoyant force. A length of timber or a floating log is an example of this condition. If, however, a weight is added at some location along its length, the equilibrium between the weight and buoyancy distribution is disturbed, and a bending moment is introduced. Throughout part of the length of the body, the upward forces of buoyancy are exceeded by the downward forces of weight, while for the remainder of the length the reverse will be true. The total buoyant force and weight must, however, remain equal for the body to remain floating in static equilibrium.

To visualize this principle one should imagine a rectangularly shaped barge, such as the one in figure 5-8, whose length is *L*. Assume that the barge has a constant weight per unit length and that weightless bulkheads and ends divide it into four equal compartments. There are no concentrated local weights on board, and the barge is floating at the indicated waterline. As shown in figure 5-8, the weight and buoyancy per unit length must coincide. The total area under the weight curve equals both the total weight and the buoyancy, because the barge is floating in static equilibrium. No bending moment is present, as the weight of the barge is supported equally throughout its length by the buoyancy.

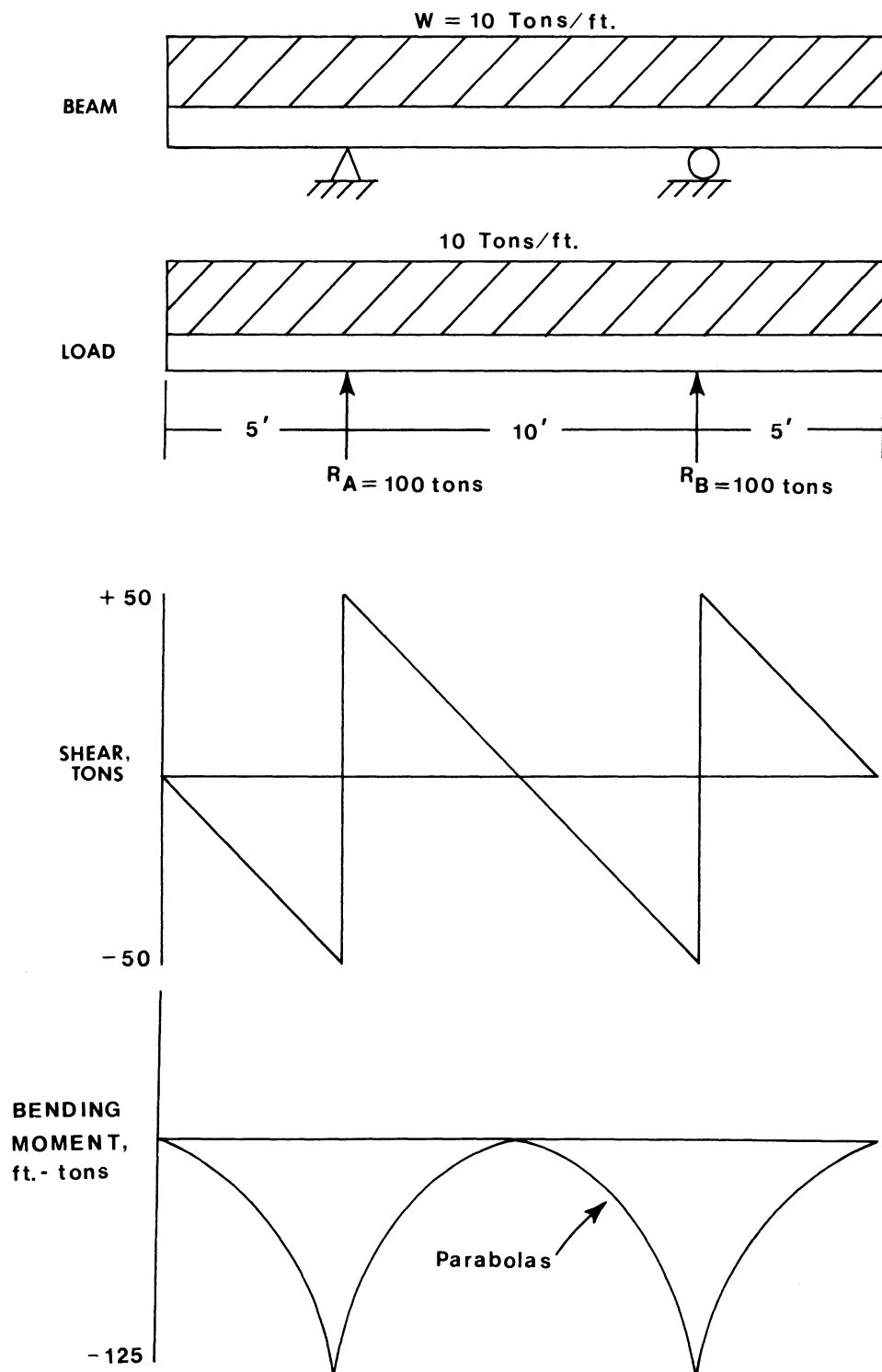


Figure 5-7. Load, shear, and moment diagrams

Now, suppose the two inner compartments, numbered 2 and 3, are filled with a cargo such as a liquid. The barge will sink to a deeper draft, increasing the buoyant force until it equals the new displacement. The added weight, however, is concentrated in two com-

partments, 2 and 3, as indicated in figure 5-9. Compartments 1 and 4 retain the same weight per unit length as in the unloaded condition. In the two end compartments, the buoyancy will exceed the weight by the same amount that the weight will exceed the buoyancy in

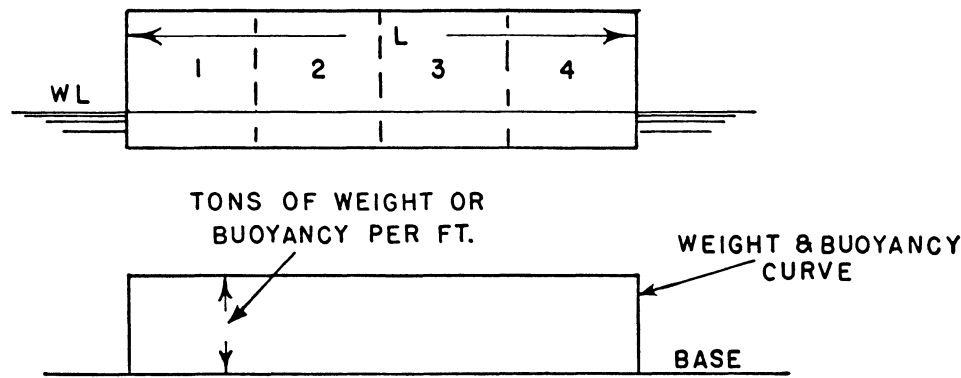


Figure 5-8. Unloaded barge

loaded compartments 2 and 3. The total weight must equal the total buoyancy in order to satisfy equilibrium conditions. Thus, the total area under each curve is the same.

The barge in this condition of loading is subjected to a bending moment throughout its length, due to the difference in weight and buoyancy distribution throughout the length. If this loading is then plotted as the difference between the weight and buoyancy for each increment of length, the result is a graphic representation of the net loading over the entire length of the barge. The load curve illustrates the net result of the difference between unit weight and unit buoyancy. In figure 5-10, the values below the base line of the load curve refer to upward forces and the values above the base line to downward forces, as indicated by the arrowheads. Note that the area of the load curve above the axis of reference must equal the area of the load curve below the axis.

The load curve may be further used to determine the vertical shear force at any point in the length of the barge. By definition, *the vertical shear at any transverse section is the algebraic sum of vertical forces to the left of the section. One may sum up the area under the load*

curve to the left of any section to give the shear at that section. Consequently, to represent the shear graphically, as in figure 5-10, plot these values of shear as ordinates on an appropriate scale at each designated section. The shear curve is the integral of the load curve. The shear at any point x_1 in the length L is

$$V_{x_1} = \int_0^{x_1} w_x dx$$

where w_x = load at any point
 dx = increment of length.

Notice that the maximum shear is exerted at the extremes of the loaded sections where the load curve passes through zero values at the base line. Also note that the total positive area under the shear curve will equal the total negative area and that shear at the ends is zero.

One may utilize the shearing-force curve to obtain the bending-moment curve. By definition, *the bending moment at any point along the length of the barge is the algebraic sum of the moments of forces to the left of that point.* Thus, since the area under the shearing-force

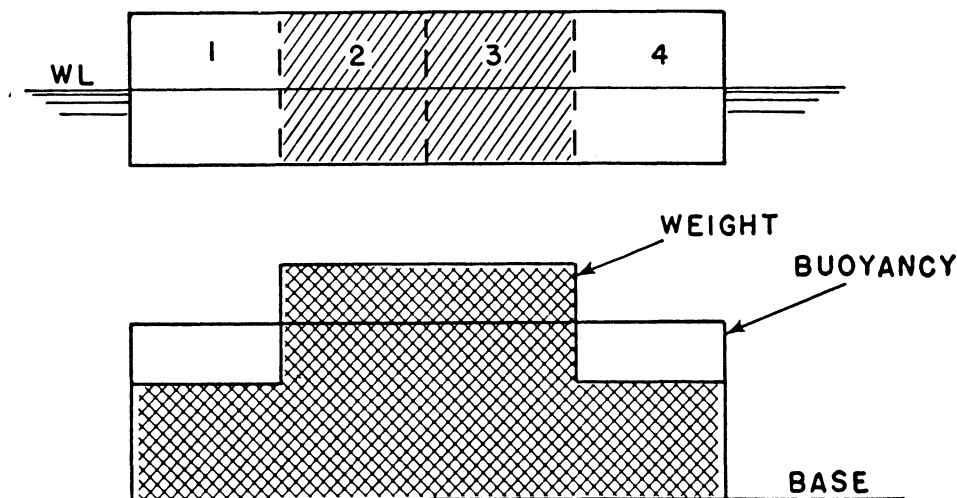


Figure 5-9. Loaded barge

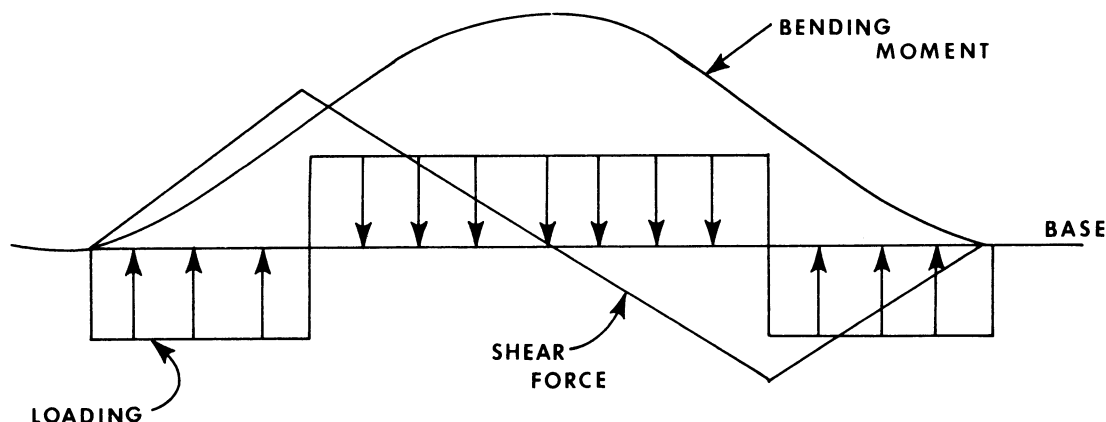


Figure 5-10. Barge strength curves

curve up to any section represents the moment of that section, the bending moment at various sections may be obtained by summing up the total area under the shear curve to the left of those sections; that is,

$$M_{x_1} = \int_0^{x_1} V_x dx$$

or

$$\int_0^{x_1} \int_0^{x_1} w_x dx dx$$

where M_{x_1} = bending moment at point x_1 in the length
 V_x = shearing force at any point.

The bending-moment curve is symmetrical in figure 5-10 because of the symmetrical loading. Note that the ordinate scales for the three curves differ. The units for the ordinate scale of the load, shearing force, and bending-moment curves are tons per foot, tons, and foot-tons respectively.

Example 5-3: Draw the load, shear, and moment diagrams for a box-shaped barge ($C_B = 1.0$), as shown in figure 5-11. The barge is 100 feet long with five equal-sized cargo tanks each 20 feet long. The beam is 35 feet, the molded depth is 20 feet, and the barge draws 10 feet of seawater as loaded. The first and last tanks contain 87.5 tons of fuel oil and the second and fourth tanks each contain 262.5 tons of fuel oil. The middle tank is empty. Assume the hull has a uniform weight distribution.

Solution:

$$\begin{aligned} \Delta_{\text{loaded}} &= \rho g \nabla = C_B L B T / 35 \text{ ft}^3/\text{ton} \\ &= \frac{100(35)10}{35} \\ &= 1,000 \text{ tons} \end{aligned}$$

$$\begin{aligned} W_{\text{cargo}} &= 2(87.5) + 2(262.5) \\ &= 700 \text{ tons} \end{aligned}$$

$$\begin{aligned} W_{\text{hull}} &= \Delta_{\text{loaded}} - W_{\text{cargo}} \\ &= 1,000 - 700 = 300 \text{ tons} \end{aligned}$$

With this information the magnitude of the distributed loads can be calculated.

$$\begin{aligned} w_{BF} &= \text{distributed buoyant force} \\ &= \frac{\Delta_{\text{loaded}}}{L} = \frac{1,000 \text{ tons}}{100 \text{ ft}} \\ &= 10 \text{ tons/ft} \end{aligned}$$

$$\begin{aligned} w_L(1) &= w_L(5) \\ &= \text{distributed load (first and fifth compts.)} \\ &= \frac{\text{wt of oil in compt.}}{\text{compt. length}} \\ &= \frac{87.5 \text{ tons}}{20 \text{ ft}} \\ &= 4.375 \text{ tons/ft} \end{aligned}$$

$$\begin{aligned} w_L(2) &= w_L(4) \\ &= \text{distributed load (second and fourth compts.)} \\ &= \frac{\text{wt of oil in compt.}}{\text{compt. length}} \\ &= \frac{262.5 \text{ tons}}{20 \text{ ft}} \\ &= 13.125 \text{ tons/ft} \end{aligned}$$

$$\begin{aligned} w_{\text{HULL}} &= \text{distributed hull weight} \\ &= \frac{W_{\text{HULL}}}{L} = \frac{300 \text{ tons}}{100 \text{ ft}} \\ &= 3 \text{ tons/ft} \end{aligned}$$

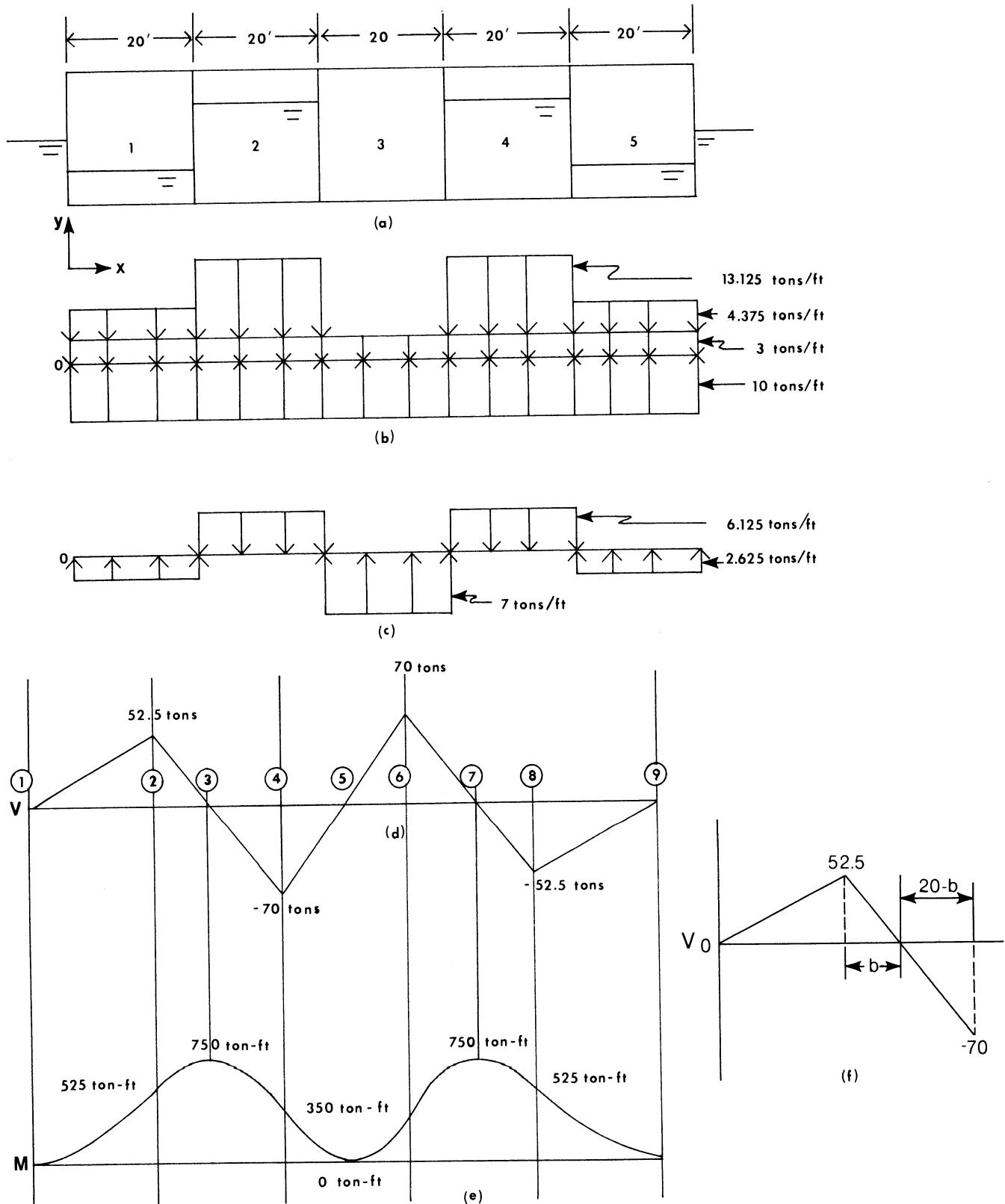


Figure 5-11. Strength curves for example problem 5-3

The distributed loads acting on the box-shaped barge are shown in the weight, buoyancy, and hull-weight curve in figure 5-11(b). By summing the various loads acting on the barge, a simplified (net) load can be drawn.

net load (first and fifth compts.)

$$\begin{aligned} &= w_{BF} - w_{HULL} - w_L(1) \\ &= 10 \text{ tons/ft} - 3 \text{ tons/ft} - 4.375 \text{ tons/ft} \\ &= 2.625 \text{ tons/ft (acting up)} \end{aligned}$$

net load (second and fourth compts.)

$$\begin{aligned} &= w_{BF} - w_{HULL} - w_L(2) \\ &= 10 \text{ tons/ft} - 3 \text{ tons/ft} - 13.125 \text{ tons/ft} \\ &= -6.125 \text{ tons/ft (acting down)} \end{aligned}$$

net load (third compt.)

$$\begin{aligned} &= w_{BF} - w_{HULL} - w_L(3) \\ &= 10 \text{ tons/ft} - 3 \text{ tons/ft} - 0 \\ &= 7 \text{ tons/ft (acting up)} \end{aligned}$$

The shear diagram is drawn by finding the area under the net-load diagram to the left of each cargo tank at the various stations.

$$\begin{aligned} V_{20} &= 2.625(20) = 52.5 \text{ tons} \\ V_{40} &= 52.5 - 6.125(20) = -70 \text{ tons} \\ V_{60} &= -70 + 7(20) = 70 \text{ tons} \\ V_{80} &= 70 - 6.125(20) = -52.5 \text{ tons} \\ V_{100} &= -52.5 + 2.625(20) = 0 \text{ tons} \end{aligned}$$

Note that for the sign convention adopted, an upward net load/ft produces a positive increase in shear when starting from the left-hand side of the net-load curve.

In order to properly determine the moment diagram, the zero crossings in the shear diagram must be located. This can be accomplished by using similar triangles, as illustrated in figure 5-11(f) for compartment 2.

Let b equal the distance from the left-hand bulkhead. Then

$$\frac{b}{52.5} = \frac{20 - b}{70}$$

$$122.5b = 1050$$

$$b = 8.57 \text{ ft from the left bulkhead.}$$

The zero crossing in compartment 4 is therefore 8.57 feet from the right bulkhead or 11.43 feet from the left bulkhead, since the shear curve to the right of midships is the inverse of that to the left. The zero crossing in the center compartment is seen to be at midships, as the loading is symmetrical about that point. (Most ships

are *not* symmetrical fore and aft. Therefore, their load, shear, and moment diagrams do not exhibit the symmetries found in this example problem.)

The bending-moment diagram is constructed by finding the area under the shear curve to the left of the section. The bending moment should be calculated for all zero crossings on the shear curve (minima or maxima on the M diagram) and for all slope sign reversals on the shear curve (inflection points on the M diagram). Note that areas below the axis on the shear diagram will decrease the value of the bending moment.

$$M_1 = \int_1^1 V dx = 0$$

$$\begin{aligned} M_2 &= \int_1^2 V dx \\ &= \frac{1}{2} (52.5 \text{ tons})(20 \text{ ft}) = 525 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_3 &= M_2 + \int_2^3 V dx \\ &= 525 + \frac{1}{2} (52.5 \text{ tons})(8.57 \text{ ft}) = 750 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_4 &= M_3 - \int_3^4 V dx \\ &= 750 - \frac{1}{2} (70 \text{ tons})(11.43 \text{ ft}) = 350 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_5 &= M_4 - \int_4^5 V dx \\ &= 350 - \frac{1}{2} (70 \text{ tons})(10 \text{ ft}) = 0 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_6 &= M_5 + \int_5^6 V dx \\ &= 0 + \frac{1}{2} (70 \text{ tons})(10 \text{ ft}) = 350 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_7 &= M_6 + \int_6^7 V dx \\ &= 350 + \frac{1}{2} (70 \text{ tons})(11.43 \text{ ft}) = 750 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_8 &= M_7 + \int_7^8 V dx \\ &= 750 - \frac{1}{2} (52.5 \text{ tons})(8.57 \text{ ft}) = 525 \text{ ton-ft} \end{aligned}$$

$$\begin{aligned} M_9 &= M_8 + \int_8^9 V dx \\ &= 525 - \frac{1}{2} (52.5 \text{ tons})(11.43 \text{ ft}) = 0 \text{ ton-ft} \end{aligned}$$

The results of these calculations are shown on figure 5-11(e). The double-positive sign convention for the integrals was chosen for ease of construction of the graphical solution. Starting from the left-hand side, positive loads yield positive slopes on the shear curve and these in turn yield positive curvature on the bending-moment curve. Negative loads yield negative slopes on the shear curve, which yield negative curvature on the bending-moment curve. A double-negative sign convention for the integrals (equations 5-1 and 5-3) will invert the shear curve, which is equivalent to integrating from the right-hand side. Regardless of the sign convention on the definition of shear, nearly all texts define positive bending moments as upward curvature. This means the bending-moment diagram is opposite to the actual deflection of the beam.

The purpose of determining the maximum values of shear and bending moment is to determine what maximum stresses are induced in the structure. The next section considers the bending or flexure stresses.

5-6 Flexure Stresses in Beams: The Neutral Axis

Flexure theory begins with several assumptions. The beam is composed of homogeneous material of uniform density, is symmetrical about a vertical plane through the longitudinal axis, and is in pure bending with no shear force present. Thus there are no axial loads and the pure bending can be considered to be provided by one couple at each end of the beam, as shown in figure 5-12. As a result of the couples at the ends, internal bending moments are generated, and the beam deforms such that plane sections remain plane and perpendicular to the longitudinal axis of the beam after bending, as illustrated in figure 5-12, where the curvature of the beam has been greatly exaggerated. Furthermore, it is assumed that Hooke's law is applicable and that the Young's modulus for the material is the same in tension and compression.

NEUTRAL AXIS

A beam of finite cross section, when subjected to a bending moment, has a plane or surface of demarcation somewhere within it, on one side of which the fibers are in tension and on the other side of which they are in compression. At any transverse section through the beam, the line of this surface of demarcation is normally referred to as the neutral axis. Consider a beam of rectangular section loaded in pure bending that deforms as illustrated in figure 5-12.

Under these circumstances, the neutral surface and neutral axis are halfway between the upper and lower surfaces of the beam. Above the neutral axis the fibers are in compression and below it they are in tension. Notice that the stresses of tension and compression increase in intensity with their distance from the neutral axis. (The actual variation of this intensity is the key to the primary problems of longitudinal ship strength and will be considered quantitatively in the subsequent discussion.)

Where the cross-section area of the beam is not rectangular (as in a T-section), the neutral axis will be at the centroidal axis of the beam. In any case, as noted above, the greater stresses occur farther from the neutral axis. This would naturally suggest a beam section, such as an I-beam, where the greater sectional area is concentrated farthest from the neutral axis. In the case of the I-beam, or any other symmetrical beam where the upper and lower flanges are identical and the web is uniform, the neutral axis is again halfway between the top and bottom. However, where the upper flange is greater than the lower flange or vice versa, the neutral

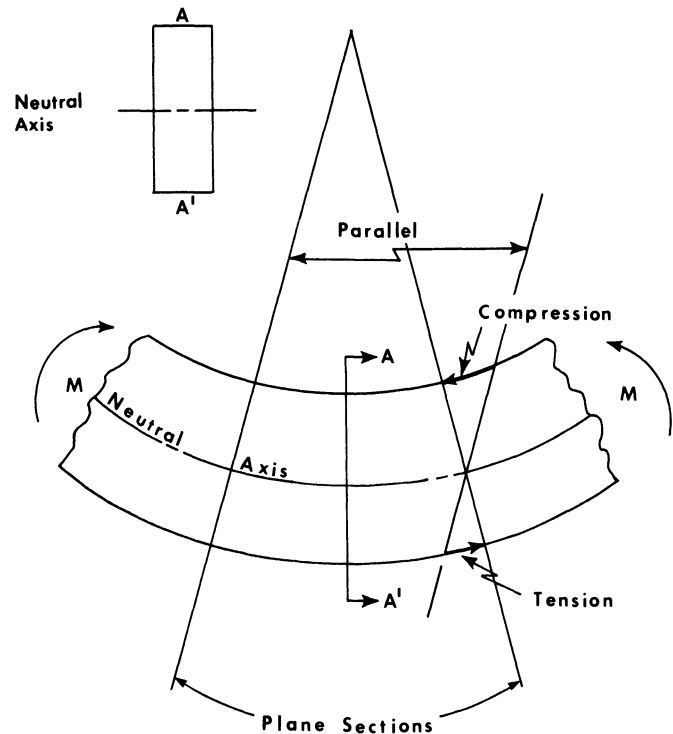


Figure 5-12. Pure bending

axis will be closer to the greater flange, the distance being inversely proportional to the distribution of area.

LINEAR VARIATION OF STRAIN AND STRESS

In 1773 the French engineer C. A. Coulomb published a paper in which he proposed that tensile and compressive stresses have the same linear distribution about the neutral axis. This is the case for simple flexure theory for deformations within the elastic limit of the material. It is not true beyond the proportional limit of the material or for nonhomogeneous structures composed of materials with different Young's modulus in various sections of the structure.

For those conditions for which Hooke's law (equation 4-11b) applies, both stress and strain vary linearly with distance from the neutral axis. The assumption in simple flexure theory that the plane sections remain plane and perpendicular to the longitudinal axis of the beam means that the strain varies linearly regardless of the variation in Young's modulus for composite material beams. Thus

$$\epsilon = \frac{\epsilon_{\max}}{c} y \quad (5-5)$$

where $\epsilon_{\max}$ = maximum strain at the farthest point from the neutral axis

c = distance to the farthest point from the neutral axis

y = distance from the neutral axis

as illustrated in figure 5-13.

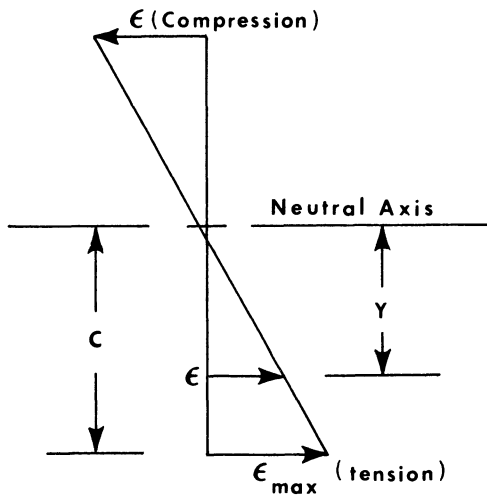


Figure 5-13. Neutral axis

For a homogeneous material loaded so the maximum stress is less than the proportional limit defined in chapter 4, the variation in axial stress with distance from the neutral axis is given by combining Hooke's law with equation 5-5.

$$\sigma = E\epsilon$$

or

$$\sigma = E \frac{\epsilon_{\max}}{c} y$$

but

$$\begin{aligned} \sigma_{\max} &= E\epsilon_{\max} \\ \sigma &= \frac{\sigma_{\max}}{c} y \end{aligned} \quad (5-6)$$

Equation 5-6 can be used to locate the neutral axis of a simple beam by summing forces in the axial direction on the free-body diagram of a section of the beam as illustrated in figure 5-14.

$$\begin{aligned} \sum F_x &= \int dF_x = 0 \\ \int dF_x &= \int \sigma_x dA \\ &= \frac{\sigma_{\max}}{c} \int y dA \\ 0 &= \frac{\sigma_{\max}}{c} \bar{y} A \end{aligned} \quad (5-7)$$

Since $\sigma_{\max}c$ and A are all nonzero, $\bar{y}$ must be zero. This means that the neutral axis of a homogeneous beam must pass through the centroid of the cross-sectional area of the beam.

ELASTIC FLEXURE FORMULA

Equation 5-6 can also be used to develop the well-known elastic flexure formula that relates axial stress to bending moment in a simple beam. Since the sum of the moments must equal zero for static equilibrium, consider the moments acting on the free-body diagram in figure 5-14. Taking moments about point A,

$$\begin{aligned} \sum M_A &= 0 \\ \int_{\text{right-hand face}} y dF - M &= 0 \\ M &= \int y dF = \int y \sigma dA \\ &= \int y \frac{\sigma_{\max}}{c} y dA \\ &= \frac{\sigma_{\max}}{c} \int y^2 dA \\ M &= \frac{\sigma_{\max}}{c} I \end{aligned} \quad (5-8)$$

or

$$\sigma_{\max} = \frac{Mc}{I} \quad (5-9)$$

but

$$\frac{\sigma_{\max}}{c} = \frac{\sigma}{y}$$

Thus,

$$\sigma = \frac{My}{I} \quad (5-10)$$

where I = moment of inertia of the cross-sectional area about the horizontal axis passing through the centroid (neutral axis)
 $= \int y^2 dA$

This equation is valid for a homogeneous beam loaded elastically with a pure bending moment applied in a plane of symmetry. Positive bending moments are associated with concave upward beam deflections. The sign convention used in developing equation 5-10 assumed that y was measured positive downwards from the neutral axis to obtain a positive axial stress (tensile stress) for positive y . When y is defined positive upwards, a negative sign is required in equation 5-10.

If axial loads and/or shear loads are present, the separately calculated local stresses may be added algebraically (superposed), provided the sum does not exceed the elastic limit of the material. Methods involving the use of Mohr's circle to determine *principal stresses* if shear loads are present are described in texts

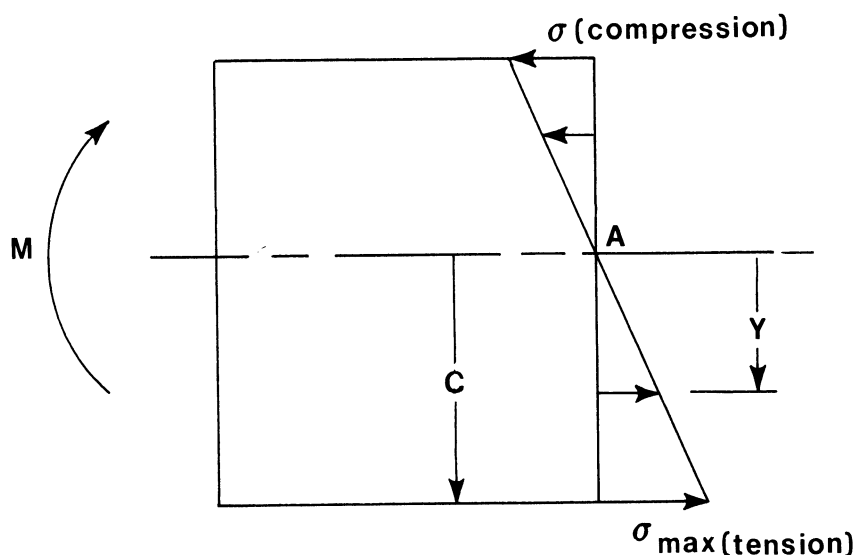


Figure 5-14. Free-body diagram of a section of a homogeneous beam

on strength of materials (for example see Popov 1968, chapter 9).

SECTION MODULUS

It can be seen from the above that the stress at any point in a beam is a function not only of its cross-sectional area but also of the distance that point is from the neutral axis. The geometric characteristics of a beam may be combined and expressed as I/c , where I is the moment of inertia of the section about the neutral axis and c is the extreme distance from the neutral axis. This ratio is generally referred to as the section modulus and can be used as a reference criterion for beams or girders of the same material in comparing their resistance to bending.

$$Z = \frac{I}{c} = \text{section modulus}$$

Thus,

$$\sigma_{\max} = \frac{M}{Z} \quad (5-11)$$

Since ships are neither simple beams nor made of homogeneous materials, some caution must be used in applying the elastic flexure formula to maximum stress calculations for ships. Although the various types of steels used in critical areas of the ship have different yield strengths, they all have essentially the same modulus of elasticity E , so the stress distributions are still linear. Those ships with aluminum superstructures are more complex to analyze, *since they have a discontinuity in the stress curve at the interface between the topmost steel deck and the aluminum superstructure*. This problem may be handled in several ways, including calcu-

lating an equivalent section modulus, assuming the material was made entirely of steel and had a linear stress distribution. The section modulus can be calculated as if the superstructure did not exist, or one can assume that the superstructure must withstand its share of the load. Such problems are discussed in *Ship Design and Construction* (chapter 7).

DISTRIBUTION OF SHEARING STRESSES IN BEAMS

In the development of the elastic flexure formula (equation 5-10), the assumption was made that only pure bending was involved and that no shearing forces were present. Thus the bending moment throughout the beam was constant and the assumption that "plane sections remain plane" was valid. When concentrated or distributed vertical loads are introduced, however, both horizontal and vertical shear stresses are generated that tend to distort the beam sections. This can be visualized by considering two identical planks loaded vertically, as shown in figure 5-15, where one plank lies on top of the other. If the planks are not glued or fastened together, the upper and lower planks will slide relative to each other, as indicated in the figure. Fastening the planks together makes the beam continuous and gives rise to horizontal shear stresses at the interconnection.

It may be shown that the magnitude of the resulting shearing stresses in beams with parallel sides is a maximum at the neutral axis. Although the presence of these shearing stresses tends to warp the plane sections, which were assumed to remain plane during elastic bending, it may also be shown that these distortions are negligibly small for slender members (that is, the length of the member is much greater than the total depth of

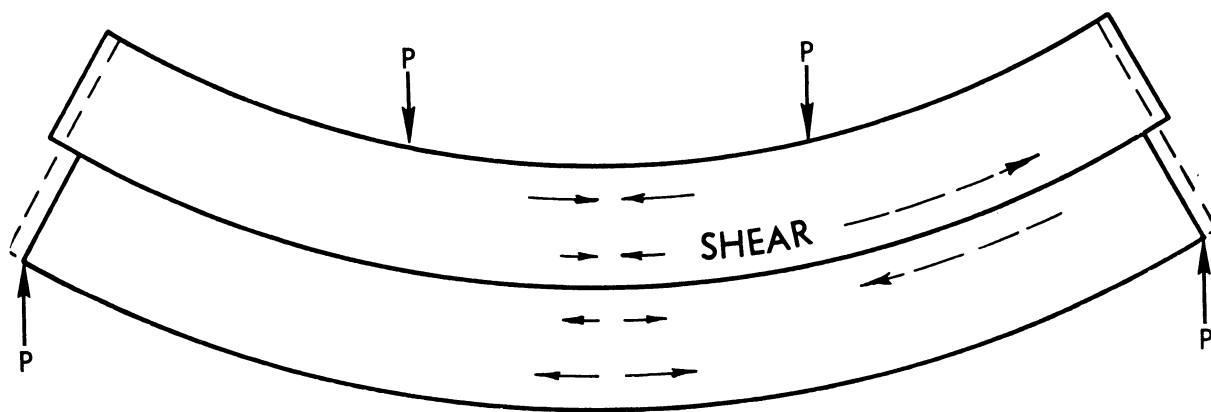


Figure 5-15. Stresses above and below neutral axis, producing horizontal shear

the member, which in turn is greater than the width). Thus, the elastic flexure formula may be used with care when concentrated loads are present.

5-7 Structural Stresses Within a Ship

The foregoing section emphasized the likeness of a ship to a single structural beam. This is, in fact, a basic premise in ship-strength calculations; however, because of the complexity of structure and the forces imposed, all of the stresses of the ship's structure must be accounted for in the design in order to check the adequacy of the ship's strength. Therefore, in order to differentiate their origin and effects, the stresses are ordinarily considered in two groups: (1) hull girder stresses and (2) local stresses.

HULL GIRDER STRESSES

A ship afloat is supported throughout by buoyant forces that vary longitudinally and transversely with the distribution of the ship's displacement or buoyant volume. These comprise the upward forces on the hull. The downward forces are the result of the distribution of all the various weights within the ship, including the weight of the ship's structure machinery, fuel, cargo, and ballast. The difference between the upward and downward forces results in a load on the ship's girder that varies throughout its length and produces an overall bending moment with the associated shear stresses.

It should be noted that the stresses resulting from a transverse bending moment are usually less severe and less important than those due to a longitudinal bending moment. In general, the size of the structural members required to give the ship adequate longitudinal and local strength will keep the transverse bending within reasonable limits.

In calculating the longitudinal strength, the simple beam theory becomes the basis of computations, and the relationship set forth previously is used. This re-

lationship for any section x in the ship's length is (from equation 5-10)

$$\sigma_x = \frac{M_x y}{I_x}$$

The members included in the calculation for the moment of inertia of any section must be continuous longitudinally (fore and aft).

At any section, the severest stresses occur in that portion of the structure most remote from the neutral axis, in the deck and bottom plating (figure 5-16). At any section, M_x and I_x are constant for a particular condition of loading for that section; however, M_x and I_x will vary from section to section along the length of the ship. The stress will therefore be maximum in the deck or bottom plating of that section where M_x/I_x is maximum (assuming that the maximum value of y will vary appreciably), usually near the midship section in most ships. The superstructure and deck houses found on destroyers (not indicated in figure 5-16) are, of course, farther from the neutral axis than the deck or bottom plating. This upper structure is often designed to be noncontinuous at intervals along the length through the use of expansion joints to prevent the structure from taking a portion of the longitudinal bending stresses.

From the discussion regarding the neutral axis of a beam, it was seen that a maximum horizontal shear stress developed along the plane of the neutral axis. Depending upon the longitudinal distribution of vertical forces (load), vertical shearing forces will develop along the ship's length. Because of the essential similarity in ship's form and distribution of load for ships with machinery amidships, this vertical shear force is maximum near the quarter-lengths from the forward and after ends, and the bending moment is maximum near midships.

A general corollary, then, for this type of ship, is that the maximum shear stresses occur in the vicinity of the neutral axis at the quarter-lengths, and the magnitude of these shear stresses often requires local

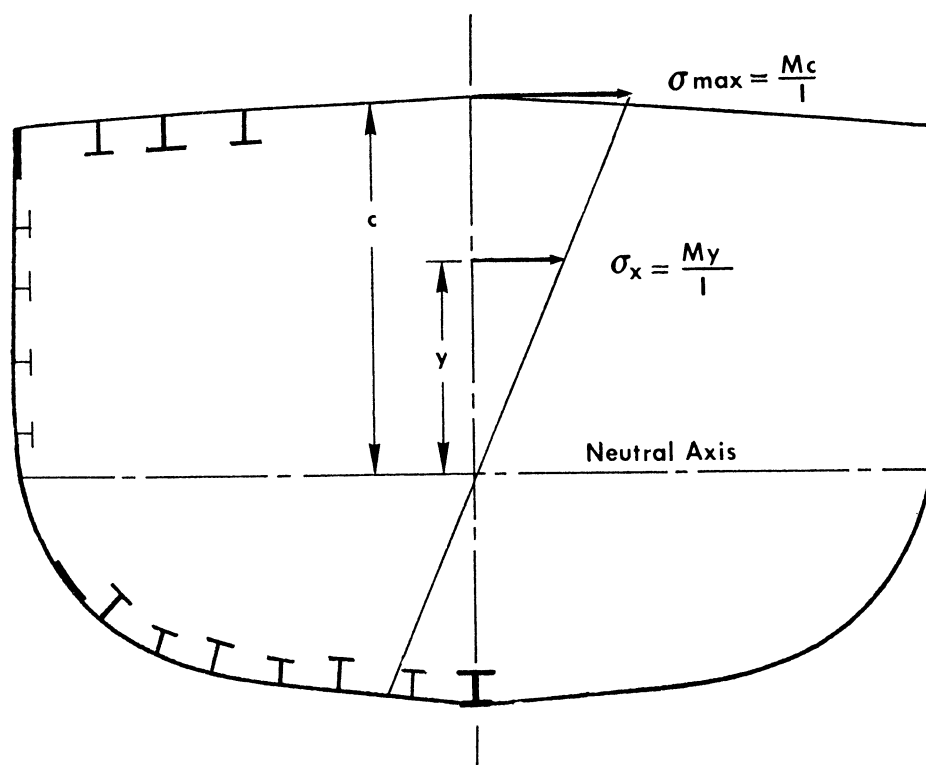


Figure 5-16. Neutral axis on a ship

strengthening at these locations in and near the side of the ship.

It should also be further stated that because of Newton's third law (for every action there is an equal and opposite reaction) and the fact that equilibrium is maintained, an opposing force equal to the shearing force is set up. This means there is a force couple at 90° to the shearing forces for both vertical and horizontal shear. In effect, the vertical shear and horizontal shear stresses are interdependent, as shown in the development of equation 4-4.

LOCAL STRESSES

Local stresses are caused by hydrostatic pressure, concentrated loads of equipment, and dynamic loading.

Each unit area of the underwater body is subjected to a water pressure proportional to its depth of immersion. The vertical component of the water pressure applied to the shell is transmitted through the internal framework and opposes the various loads of the ship. Although the horizontal components of the water pressure acting on each side cancel each other, preventing an athwartships movement of the ship, the force of the horizontal pressure components still acts on the shell.

The hull and its internal framing must resist the tendency of the water to crush the hull. When the skin of the ship is ruptured and flooding follows, the hydrostatic pressure formerly exerted on the shell is placed on the internal boundaries of the flooded space. These internal boundaries must be stiffened sufficiently to prevent their failure and hence to confine the flooding. Hydrostatic pressure is also imposed on the boundaries of intact fuel and water tanks.

The weight of each object rests at some point in the ship. These loads must be transmitted downward through the internal structure to the shell, where they are opposed by the vertical component of the hydrostatic pressure. To prevent the excessive stress of concentrated loads, extensive foundation support is used to distribute the load over a large area.

In addition to the local stresses imposed by static loads, the ship's structure may be subjected to the buffeting action of the wind, waves, liquid load, and on naval ships, the blasts of missiles and torpedo and mine explosions.

Damage to the structure will impose greater stresses on undamaged members, not only because of reduced effective cross section of the members, but also because of the discontinuities in the structure that may result in stress concentrations.

5-8 Means of Determining Ship's Strength Curves

STILL WATER

Weight calculation for use in strength curves, as well as displacement, stability, and other considerations, is begun in the early stages of design. For purposes of determining the strength curves, both the magnitude and the location of the weights must be accurately made. It will be advantageous to discuss the general practice followed in such weight calculations.

Weight Groups. To proceed with the weight calculations in an orderly fashion, the weights are classified and subdivided into groups. In naval designs, the current practice is to classify all of the component weights of the ship, its equipment, and its complement into seven primary weight groups as follows:

- Group (1)—Hull structure
- Group (2)—Propulsion plant
- Group (3)—Electric plant
- Group (4)—Command and surveillance
- Group (5)—Auxiliary systems
- Group (6)—Outfit and furnishings
- Group (7)—Armament

The above weight groups are now standard in naval-ship weight calculations, and under each of these main groups are further detailed subdivisions. The complete description of the detailed subgrouping will be found in *Ship Work Breakdown Structure*.

Weight Calculations. In the early design stage, weights are estimated by comparisons with corresponding weight groups on similar existing ships. Main structural items are laid out, and the weight is calculated directly. The weights are recalculated in much more detail in the subsequent preliminary and contract design stages and in still more detail in the detail design made at the building yard.

The weights of such standard articles as guns, pumps, and anchors are relatively easy to determine—either by reference to the manufacturer's records or the article's specifications. Other parts, such as frames, hull plating, and bulkheads, must be calculated item by item from the ship's plans. The weight of items such as turbines, boilers, and other large units must be broken down by component parts and determined from the manufacturer's specifications.

The first ship of a class to be built is frequently weighed in component form; that is, each piece of material and equipment put aboard is actually placed on a scale and its weight recorded. Later the components are totaled to give the final weight check.

The weights of the various groups are summarized, along with their vertical and longitudinal moment arms, about the keel and midship section respectively. It is

from this summarization that the weight data per unit length is determined and plotted as the weight curve. Figure 5-17 demonstrates that the weight curve is an extremely irregular one, reflecting the concentrations and nonuniformity of the ship's weight distribution.

Buoyancy Calculations. The calculations for buoyancy are relatively simple compared to those for weight. These values are determined by computing the sectional areas below the waterlines corresponding to the several loading conditions for which weight calculations are made. For any given waterline or displacement condition, these sectional areas are plotted as ordinates to some convenient scale throughout the length of the ship. A smooth curve is faired through these ordinate points, resulting in a curve that describes the buoyancy distribution longitudinally along the ship. Figure 5-17 shows that the buoyancy curve is gradual and continuous with its maximum values near the midship section.

Load, Shear, and Bending-Moment Curves. After the weight and buoyancy curves have been determined, the subsequent procedure is straightforward. The net-load curve is obtained by merely subtracting the buoyancy values from the weight values at selected intervals along the length and plotting the resulting differences at each location. This curve is then integrated by approximate means to obtain the shear curve, which in turn is integrated in a similar manner to obtain the bending-moment curve. For cargo-carrying ships, shear and bending-moment curves for various loading conditions are calculated to determine those conditions of loading that must be avoided for structural reasons. The principles involved in the actual procedure of obtaining the ship's strength curves are obviously identical to those for the simple loaded barge described previously. The meaning is also identical; however, because of the complexities of weight distribution and ship form, the analysis is tedious (if done by hand) and the irregular curves reveal a more complex strength problem. This becomes increasingly apparent when one considers the condition of the ship no longer in still water but at sea among waves.

SHIP IN A SEAWAY

The foregoing discussion considered the vessel floating in still water. This condition provides a distribution of buoyancy along the length of the vessel that is the same as the distribution of displacement below the designer's waterline. A ship among waves is alternately supported by the wave crest at the midship region or the ends; that is, the buoyancy distribution varies with the position of the wave crest passing under it. For the purpose of studying the stresses on the ship's girder among waves,

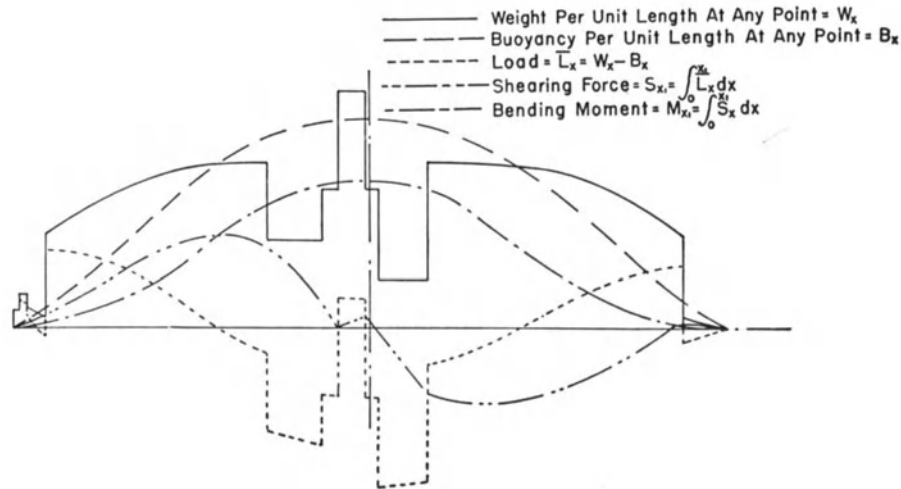


Figure 5-17. Strength curves for cargo vessel

certain values of a wave's size must be assigned in order that the corresponding buoyancy distribution may be calculated. One quasi-static method for such a study is to assume that the ship is supported by a *trochoidal* wave, whose length is equal to the length between the ship's perpendiculars and whose height is $1.1\sqrt{L}$. The evaluation of the buoyancy distributions, in one case where the wave crest is amidships with the wave hollows at the ends of the ship, and in another case where the wave hollow is amidships with the wave crests at the ends of the ship, is assumed to approximate the most severe stresses to which the ship normally will be exposed.

With a wave of these dimensions, curves showing the distribution of buoyancy are drawn. Figures 5-18 and 5-19 indicate the variation in the buoyancy curves for these conditions. As stated at the beginning of the chapter, when the bow and stern are riding on wave crests and the midship region is in the trough, the ship will bend with compression in the weather deck and tension in the bottom plating. The ship is sagging. When the ship advances one-half a wave length from the sagging condition, so that the wave crest is amidships and the bow and stern are in the trough, the ship will bend with the weather deck in tension and the bottom plating in compression. In this condition the ship is hogging.

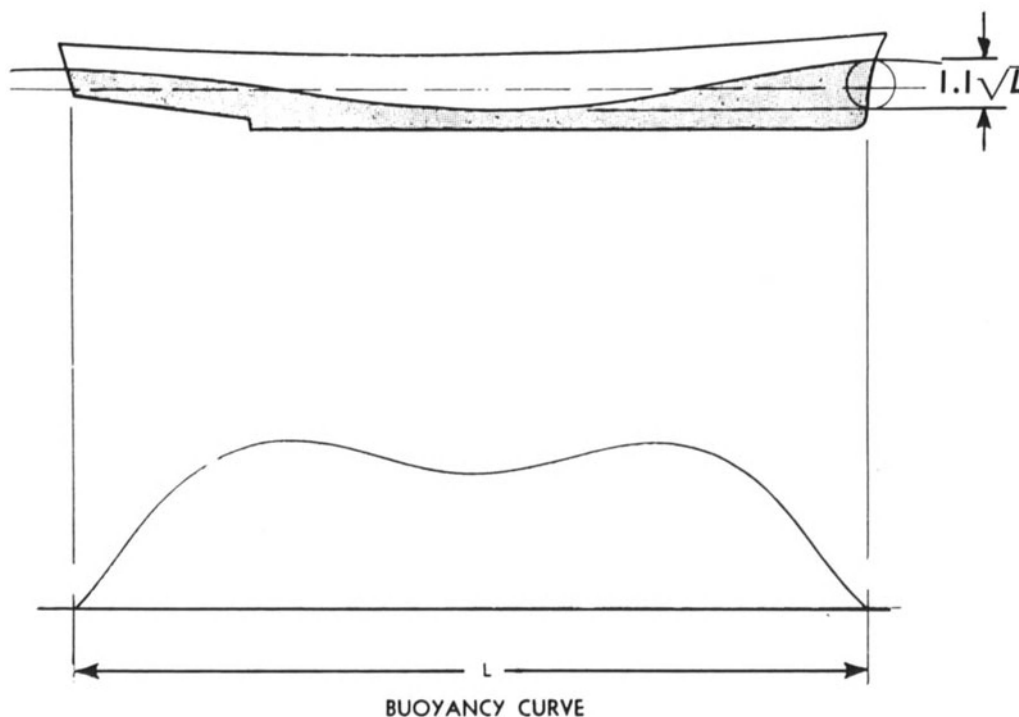


Figure 5-18. Typical sagging condition and buoyancy curve

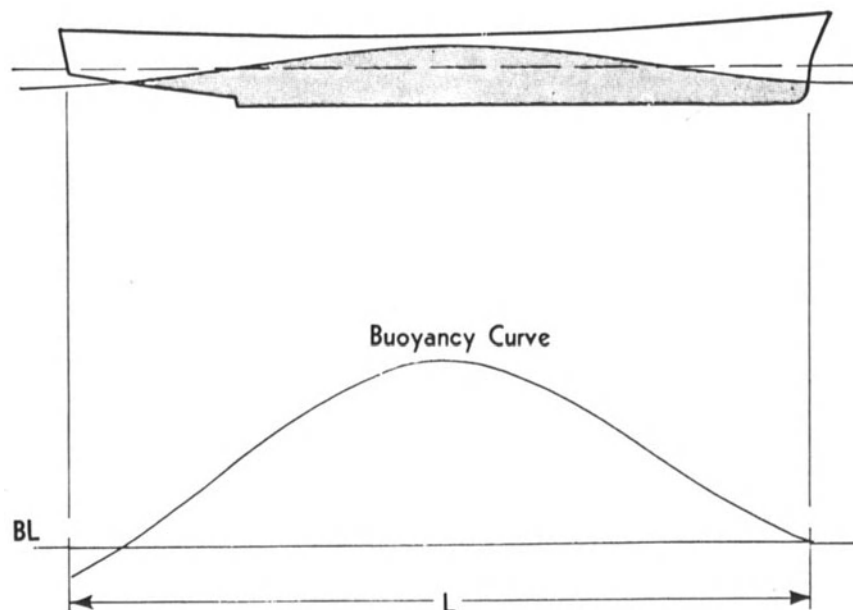


Figure 5-19. Typical hogging condition and buoyancy curve

Strength curves are normally developed for several displacements, considering both the hogging and sagging condition at each displacement. To insure that the most severe condition is examined, weight-distribution curves are constructed for the hogging condition, considering that the consumable loads amidship are expended, and for the sagging condition, considering the consumable loads at the ends are expended.

From the structural section plans, similar to those in figure 5-20 but in much more detail, the designer calculates the section modulus (I_x/C) at various sections. Knowing the bending moment at each section for each loading condition, the maximum stress at each section can be determined from the equation

$$\sigma_x = \frac{M_x y}{I_x}$$

to insure that it is within allowable limits. The maximum longitudinal bending stress occurs normally in the deck or bottom plating in the vicinity of the midship section.

To resist longitudinal bending stresses, the hull girder of figure 5-20 is composed of the deck plating, shear strakes, and deck girders at the top and the bottom plating, keel, and longitudinals at the bottom. These correspond to the flanges of an I-beam. The relatively thin side plating in the vicinity of the neutral axis acts as the web of an I-beam.

The size of each structural member must be proportionate to the load it carries. It is good design practice to avoid having any member of greater strength and weight than necessary. In such a case, the neutral axis is moved toward the stronger member, and the stress

in the extreme fiber is proportionately increased. It is also good design practice to reduce the size of structural members beyond the middle half-length of the ship. This reduction is made as gradually as possible, since the standard structural members are of constant cross section.

The quasi-static method of predicting sea loads has been supplemented by computer-based methods that use measured or assumed statistical representations of the sea (see chapter 13) to compute the loads, shear, and bending moments based on so-called *strip theory* (Salvesen, Tuck, and Faltinsen 1970). Strip theory assumes that the ship response is a linear function of wave height, which is borne out by model tests in moderate seas, but is known to be an inaccurate description of ship motions and loads in severe or extreme seas. In order to measure the bending moments actually experienced in sea conditions along various ocean routes, the American Bureau of Shipping (ABS) and other international agencies have sponsored the instrumentation of various ships, including the 312,000-dead-weight-ton tanker, the *Universe Ireland*, with strain gauges, electronic recorders, and data-processing equipment to determine the actual stresses experienced by the hull in a seaway. Tests using segmented models to measure the local bending moments between various sections of the hull, induced by simulated ocean waves in a ship-model towing tank, have been used to verify the accuracy of the computer predictions based on strip theory. This is a rapidly changing part of the ship-design process, with a number of computer programs available and under development as design tools for the naval architect.

RELATION AMONG STRENGTH CURVES

In all strength curves the following general relations exist:

1. The area under the buoyancy curve must equal the area under the weight curve.
2. The centroids of the weight and buoyancy curves are vertically in line.
3. The net area of the load curve must be equal to zero.
4. Peak shear values will exist at those points in the length where the load curve crosses the axis. (The word *maximum* is not used, because there are usually several peak values.)
5. Maximum bending moments occur where the ordinate of the shear curve is zero.
6. Points of inflection occur in the bending-moment curve where the ordinates of the shear curve have peak values (where the load curve intersects the base line).
7. The shearing force and bending-moment curves have zero values at their extremities.

CONTINUITY IN STRENGTH

The effectiveness of the ship's hull as a box girder and the effective application of the beam theory equation depend primarily on the longitudinal continuity of the strength members in the hull structure. Unfortunately, because of operational requirements, the hull structure must contain many openings in the vicinity of which it is impossible to maintain the continuity of some structural members or structural plating. To as great an extent as possible, these openings are located in such a way as to eliminate interference with principal structural members. For example, it is prohibited in any ship to cut, notch, or place an opening of any kind in the main-deck stringer plates. Likewise, the keel and other key longitudinal members cannot be disrupted by an opening or any other part of the structure that might interfere with their continuous nature (see figure 5-20). However, because the deck, side, and bottom plating make up a considerable portion of the longitudinal strength, it becomes a problem of design to maintain as much continuity as possible and still allow for nec-

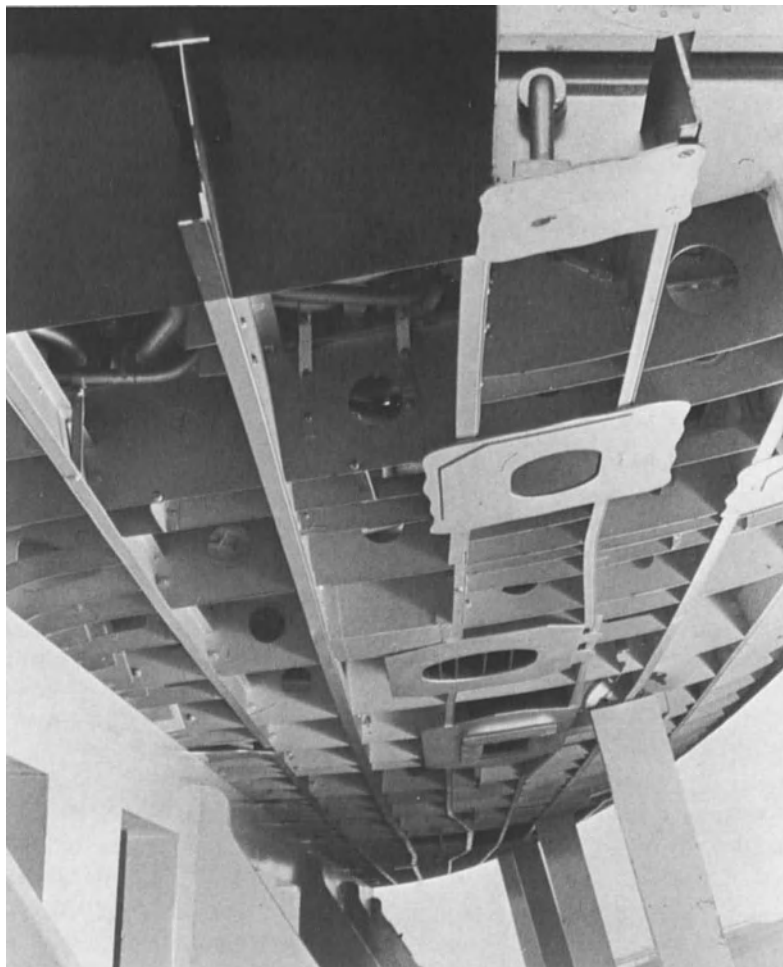


Figure 5-20. Sectional model of a destroyer, showing continuity of longitudinals

essary openings, such as propulsion-plant intake and exhaust ducting and sea chests for condenser cooling water.

An opening in the shell of the hull will reduce the section modulus in that location because of the sectional area of plating lost by the opening. One method of compensation for the lost area is to reinforce the periphery of the opening with additional plating, called *doubler plates*, or thicker insert plates. These doubler plates, as a general rule, are of such dimension that they are equivalent to the area of the opening. The shape of the opening is also of critical importance. Any sharp corners or square cuts around the hole must be avoided because of the concentration of stress at these points of abrupt change in direction. Not only are openings involved in this phenomenon of stress concentration, but also any feature of design that might result in an abrupt change in curvature or direction of plating. This is noted in ships where superstructure decks or raised forecastle decks become a part of the strength deck. In such design, it is necessary to fair the side plating from one deck level to another in a gradual, smooth curve to eliminate the angle.

Investigations have been conducted by the U.S. Maritime Commission to determine adequate hatch-corner design. Failure of deck plating in large merchant ships in the vicinity of hatch corners, because of great stress concentration, has resulted in new construction and alterations, which produce a rectangular hatch with rounded corners not only in the deck plating but also in the hatch framing. The framing and coaming around the hatch have become much stronger structures in modern design, resulting in an all-welded, integral unit.

5-9 Structure

Because of the varied types of ship and craft in naval and maritime service, it will be necessary to discuss a ship's structure in general terms. In the following development, no attempt will be made to examine all types of structure, but rather typical practices will be emphasized. The requirements for strength, resistance to battle damage and flooding, and flexibility in ship control all demand that a naval vessel be a more complex structure than a merchant ship. Not only must the structural members be suited to greater stresses and be able to stand up when adjacent members have been damaged or destroyed, but the entire vessel must also be subdivided more extensively to restrict and contain flooding damage. Because of the scope of this text and the objective of the student, the material related here to the structure of ships must, in any event, be limited and elementary in nature. For the student or reader seeking more detailed information there are numerous ship-building texts and manuals available, such as the previously mentioned *Ship Design and Construction*.

THE HULL STRUCTURE

The hull of a ship must be primarily a buoyant, stable container for all of the machinery, living facilities, equipment, and cargo spaces required for its functioning as an integral unit. It must also be of adequate strength and seaworthiness and have an elongated shape of faired, smooth, underwater form for minimum resistance. The hull must have ample buoyancy and a form that provides stability.

It is possible in describing the hull to liken it to a box-shaped girder. The most significant modifications of this box-like form are the tapered form of the sides and bottom at both ends and the sheer of the deck. Because of the rounding off at the bilges and the fairing in of the bottom toward the ends to provide a smooth underwater form, the box-like form of the hull is further modified. Also, because the streamlined underwater form is symmetrical about the longitudinal centerplane, it is necessary to provide a rigid centerplane girder from end to end, as illustrated in figure 5-21. This girder, the *keel structure*, is one of the most important structural parts of the ship. Keeping in mind this centered structural member, the structural frame is further composed of longitudinal and transverse members integrally welded together, forming a framework to which the watertight skin or plating is attached. This framework will be referred to as the *structural frame* of the ship. It is made up of longitudinal and transverse members—beams, girders, plates, bars, etc.—that run in planes fore and aft and athwartships.

In addition to the structural frame, a considerable portion of the ship's overall strength and rigidity is provided by the watertight shell or plating. This plating, an integral part of the hull structure, closes in the whole box-like girder, including the bottom, sides, and weather deck.

With the addition of transverse bulkheads to provide transverse rigidity and watertight subdivision, the hull structure becomes the integral unit used to meet the requirements set forth in the beginning of this section.

It should be emphasized here again that the hull's strength depends upon a uniform and continuous structure. Any break or rupture in the structural frame or in a portion of the shell plating results in a discontinuity of the structure with a consequent concentrating of stresses in adjacent members.

THE STRUCTURAL FRAMING

Attached to the keel structure transversely are members called *frames*. The frames, in general, run continuously from the keel to the main deck edge. Distinguished from the frames, although actually a continuous part of them, are the *floors* in the ship's bottom. The floors are actually deeper frames that run from the keel out to the

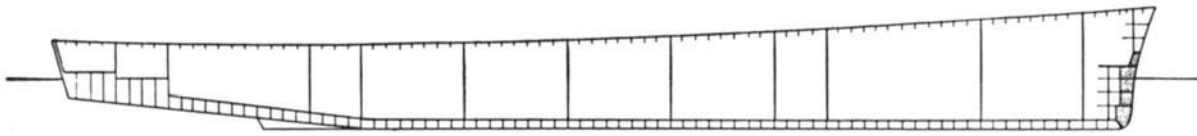


Figure 5-21. The longitudinal center girder of a naval vessel

turn of the bilge or to a point where a transverse horizontal line from the top of the inner vertical keel meets the side of the ship. At the outer extremity of the floors, the frames are attached and run up the side to the deck line. The floors and frames together make up the major portion of the transverse structure of the ship's structural frame.

Longitudinal girders running parallel with the keel and intersecting the floors at right angles are known simply as *longitudinals*. The longitudinals and the floors divide the ship's bottom into relatively small rectangular tanks and voids collectively called the *double bottoms* (some smaller vessels lack inner bottom plating and thus have *single bottoms*). The bottom structure is thus an extensively subdivided portion of the ship, and in most ships it is the strongest flange of the whole ship's girder. The longitudinals run throughout the length of the ship, continuing at regularly spaced intervals around the bilge and up the sides. The longitudinals along the sides, commonly called *stringers*, are proportionately lighter than those along the bottom, as are the frames. The

arrangement of these in a destroyer may be seen in figures 5-20 and 5-22.

The hull is closed in on the top by the deck structure, allowing for appropriate openings and portions of the ship that extend through the deck. The transverse members of the deck-framing structure are known as *deck beams*. The lighter deck beams in naval vessels are ordinarily spaced at regular intervals with deep deck beams, particularly where there are openings and discontinuities in the deck. The longitudinal members of the deck-framing structure are called *deck girders*, with the heaviest girders located, insofar as possible, at the center and near the outboard edges. The dimensions of frames, girders, plating, etc. are commonly called *scantlings*.

Complete transverse bulkheads, placed at appropriate locations, extend from keel to deck and side to side and provide considerable transverse stiffening as well as watertight subdivision. In addition to the frames and bulkheads used as transverse strength members, deeper frames and built-up frames called *web frames* are found

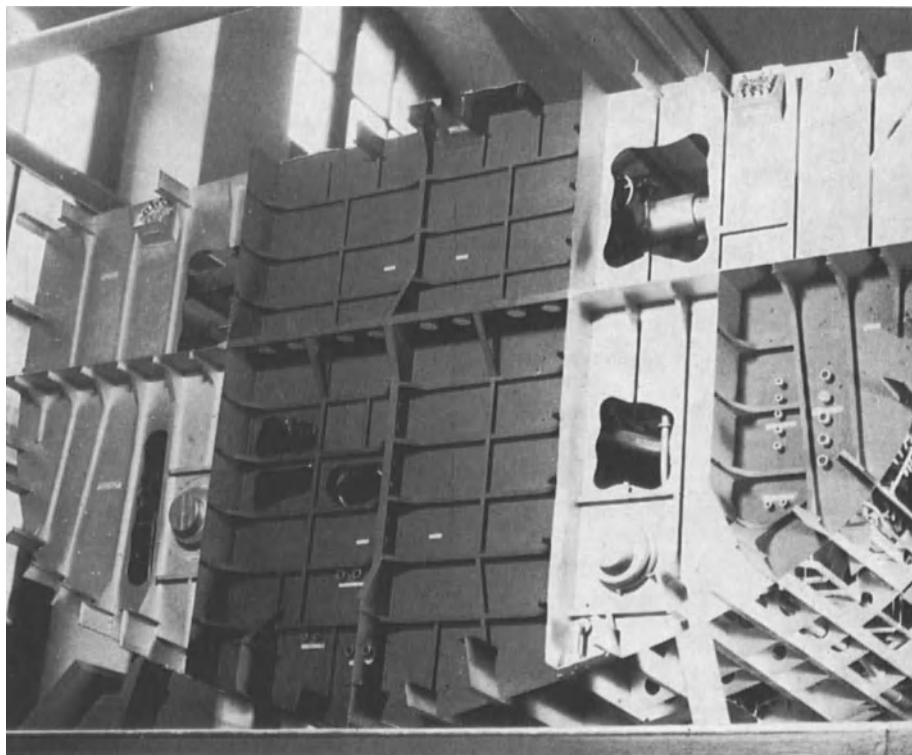


Figure 5-22. Sectional model of destroyer, showing structure. (Note ends of longitudinals and stringers. Portions of floors and frames are shown in lower right.)

in many modern ships. Web frames are most adaptable to combatant vessels, because they provide excellent compartmentation for protective layer tanks. Figure 5-23 shows the deep web frames adjacent to the machinery space in a destroyer. In this case, they provide the boundaries of the feedwater tanks and are part of the wing tank system.

There are two distinct systems of framing in common use that are distinguished from each other by the terms *transverse* and *longitudinal*. The transverse system consists of closely spaced, continuous frames with widely spaced, intercostal (separate), deep longitudinals. The longitudinal system consists of frequently spaced but shallower longitudinals with widely spaced intercostal web frames. In merchant shipbuilding practice, one or the other of these systems is used in nearly pure form, but modern naval surface ships use a combination of the two systems, which gives greater continuity of structure and less weight for a given strength. Figures 5-22 and 5-23 are examples of this combination system. Figure 5-22 shows the deep and shallow longitudinals, stringers, and deck girders piercing a main transverse bulkhead. The web frames in figure 5-23 are fitted every three or four frame spaces between lighter frames.

It should be emphasized at this point that the framing practices in combatant naval vessels differ from those in noncombatant and merchant vessels to such an extent that it is difficult to identify the various types by a

rigorous nomenclature. Primarily because of greater strength requirements and built-in resistance to damage, naval vessels have generally deeper frames. The primary frames in a destroyer, for example, could all be called web frames, since the simple T-bar frames are only used in the extreme bow sections for side stiffening (see figure 5-24) and are spaced more closely than other frames. Adjacent to machinery spaces, the frames are so increased in depth that they might be called deep web frames, bulkhead frames, or compartment frames. Some of them have large lightening holes, and others form a continuous ring inside a side compartment with stiffening flanges welded around their inner edges. Regardless of the shape and extent of the frames, their purpose is essentially the same in all vessels—to provide transverse stiffening, strength, and resistance to transverse forces. They supply a measure of overall longitudinal strength in their vertical and transverse support of the sides.

THE KEEL AND BOTTOM

The keel structure in most steel vessels consists of a center vertical keel and an outer flat-plate keel.

The center vertical keel in small vessels is generally a welded assembly in the form of an I-beam or T-bar. In many large combatant ships with a bottom protective layer, the center vertical keel is a vertical assembly of two or more sections, as would be formed by two I-

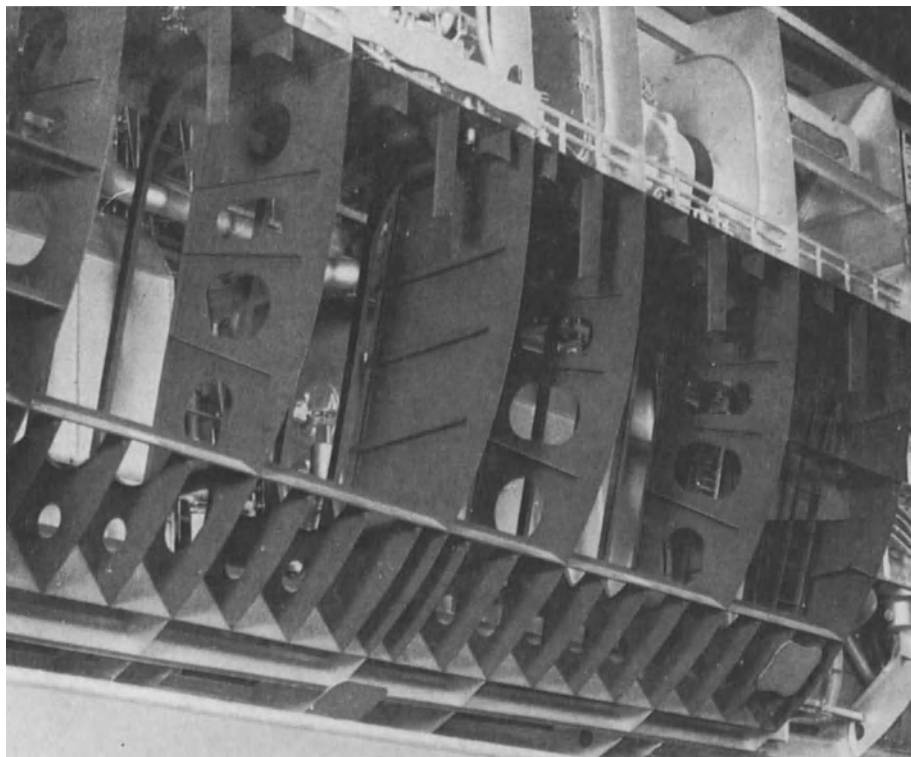


Figure 5-23. Deep frame construction in a destroyer. Web frames such as this form wing tanks for the protection of machinery spaces and storage of fresh and feed water.

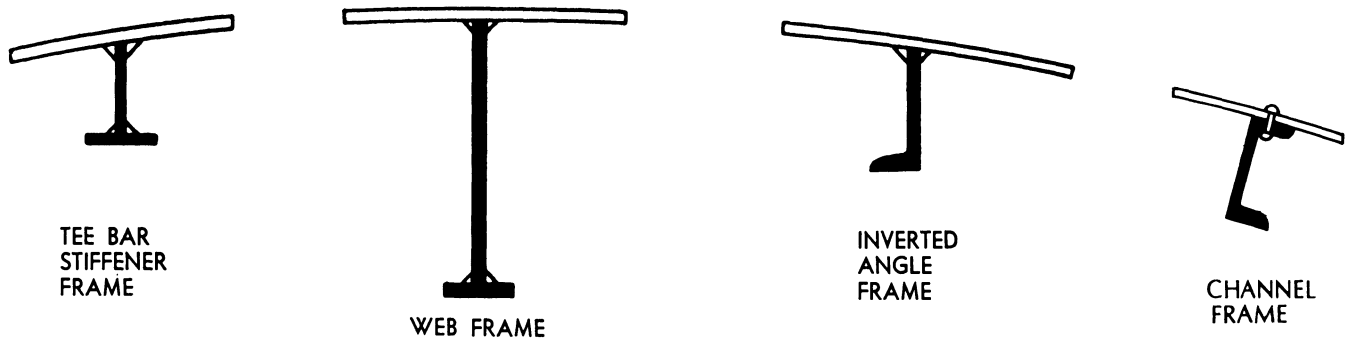


Figure 5-24. Types of simple frames (sections)

beams arranged one on top of the other. This structure is generally of welded construction composed of flat plate and angle bars. Figure 5-25 shows a typical example of a welded keel structure.

Longitudinally, the entire keel member is made up of a number of sections. These sections are seldom more than 100 feet in length.

The outer flat keel in small vessels often forms the lower flange of the center vertical keel, but in larger ships it is an additional member attached to the lower flange of the center vertical keel. This member is generally wider than either the upper or lower flange of the center vertical keel. It is not generally flat but, where necessary, conforms to the shape of the hull. This is always necessary in the vicinity of sections that have any degree of deadrise. Although the outer flat keel is of heavier gauge plate than side and bottom plating, in reality it functions as the primary, centrally located strake of the bottom plating. The line of separation between the top of this outer flat keel plate and the lower flange of the center vertical keel is the base line defined in chapter 3.

A cellular construction in a ship's bottom is the result of a large number of floors intersecting the longitudinals (see figure 5-26). These bottom compartments become liquid stowage tanks and voids, and when they are covered over by plating, an inner and outer bottom is formed that constitutes the strongest portion of the ship. The greater cross-sectional area of longitudinally continuous structure of the ship's bottom keeps the neutral axis below the midpoint in the ship's depth. The double bottom should, on large ships, be considered the integral part of the bottom strength of the ship.

In regard to the complete bottom structure, it should not be overlooked that bilge keels may contribute to the strength and structural continuity. An exception to this rule is the case where the bilge keels are not attached to the hull by a continuous weldment. However, the attachment is often continuous, and in such cases the bilge keels are similar to stringers or longitudinals. Ships' officers should remember that the bilge keels are

particularly vulnerable to damage because they are external structures. For this reason, bilge keels are usually of lighter scantlings than the shell to which they are attached, so that minor damage to the bilge keels will not result in tearing of the watertight shell.

STERN ASSEMBLY

At the after extremity of the ship is the stern assembly. Because of the great variety of types and shapes of sterns, this assembly is equally varied in its manner of construction. Generally speaking, for single-screw vessels, the stern is constructed to accommodate the propeller shaft and rudder stock bosses, as illustrated in figure 5-27. It is sometimes made up of castings and forgings, with the bosses and rudder gudgeons faired in as an integral part of the form. The stern post as such is difficult to define in modern transom sterns and multiple-screw warships, particularly those with twin rudders. Actually, the older type of stern post in such ships has been replaced by an equivalent structure of deep

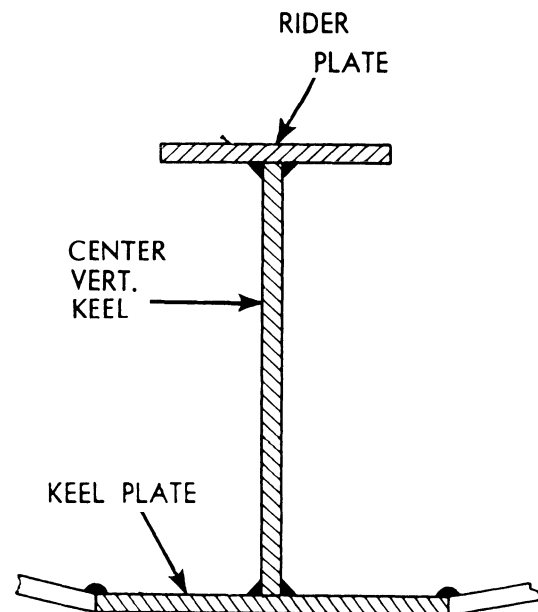


Figure 5-25. Typical welded keel section

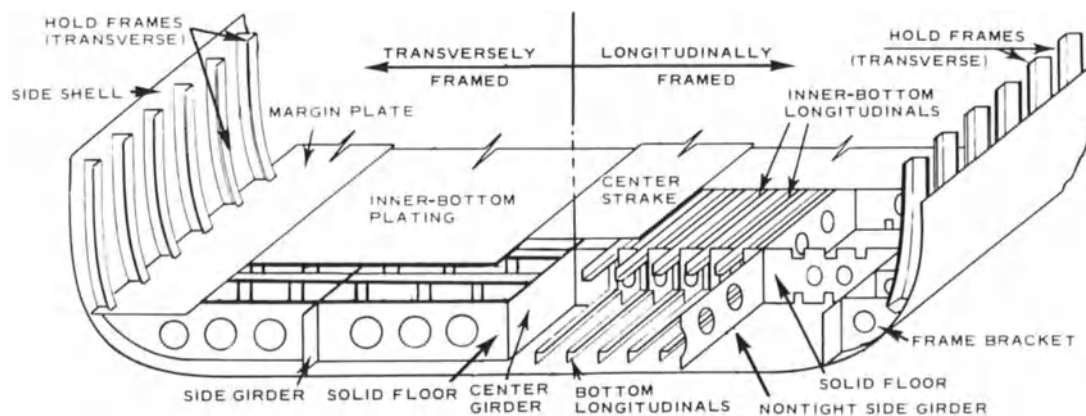


Figure 5-26. Floor with longitudinals (bottom structure). (From Taggart 1980.)

framing, both longitudinal and transverse. This structure is extended throughout the width of the bottom in the vicinity of the stern and is so constructed to accommodate the necessary rudder posts (see figure 5-28). In the vicinity of the rudder posts, the structure is strengthened to withstand the additional static and dynamic loads imposed by the rudders. The structure there, commonly known as the *rudder-post weldment*, is heat treated as a unit to relieve residual welding stresses before installation in the ship. This is a particularly distinctive feature in naval-vessel construction and is primarily the result of the need to absorb large rudder forces and

vibrations generated at high speed (figures 5-28 and 5-29).

STEM AND BOW STRUCTURE

At the forward extremity of the keel, attached integrally, is the stem assembly. Ships' bows are of varied forms, as the requirements of resistance and seakeeping dictate the shape. A form essentially bulbous at the forefoot (near the lower portion of the stem), tapering up to a sharp entrance near the waterline, and again widening above the waterline is the most common shape of the stem structure. Internally this structure has a

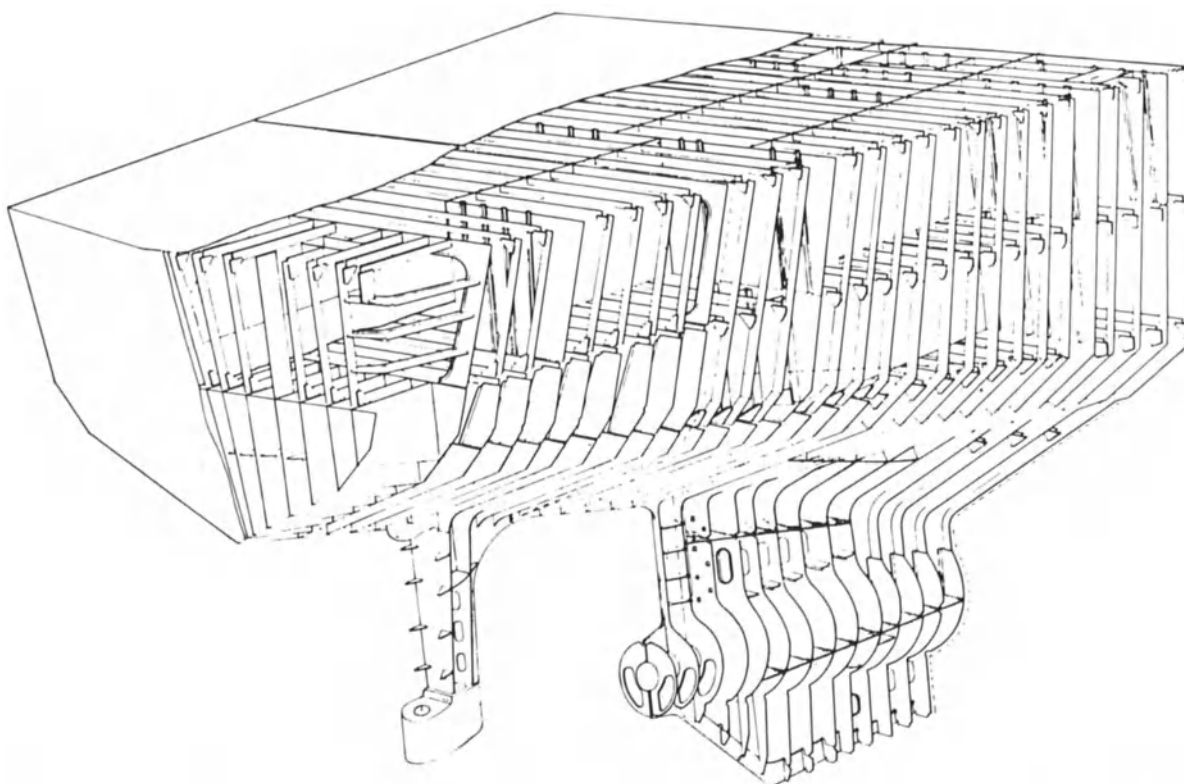


Figure 5-27. Structural frame of afterbody of a single-screw cargo ship, showing typical rudder horn and frames for propeller bossing

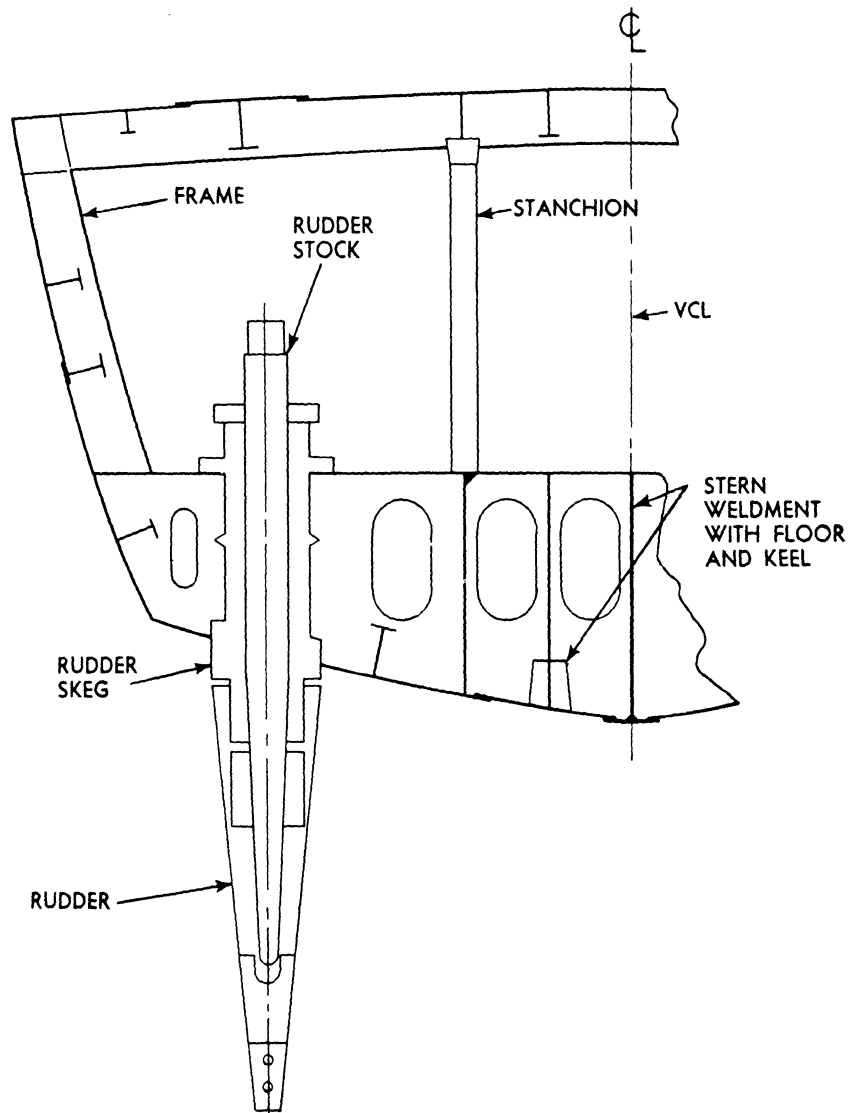


Figure 5-28. Floor and frame weldment in after section at the rudder posts (twin rudder mounts)

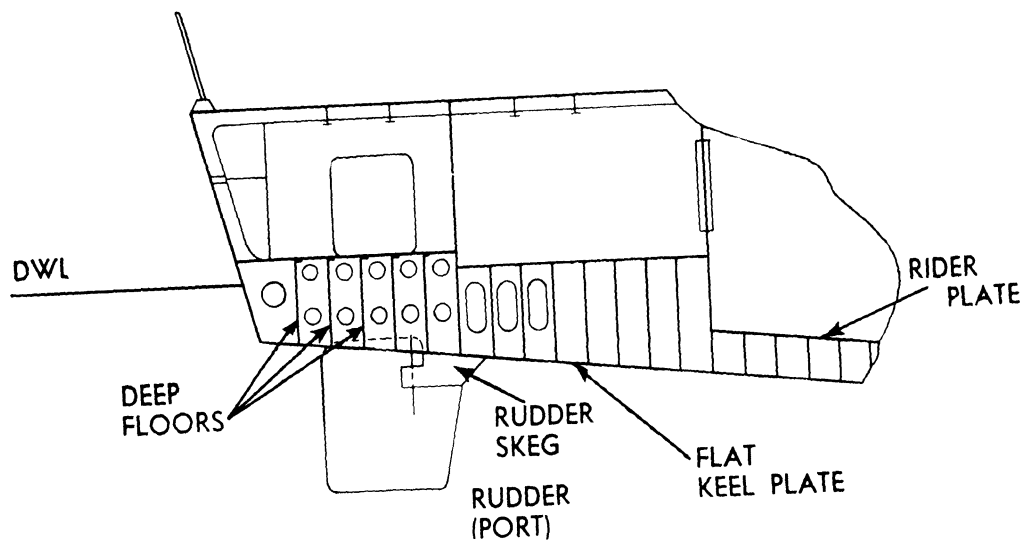


Figure 5-29. Stern and rudder weldment assembly

heavy centerline member that generally comprises the lower half of the stem. At various levels and regular intervals along the stem structure between keel and deck are horizontal members called *breast hooks*. Depending on the shape of the bow at any level, these members, made of heavy plate, are basically triangular in shape. Deep transverse framing and transverse bulkheads complete the stem assembly. The stem itself is fabricated from castings, forgings, and heavy plate, or in the case of smaller ships, heavy, precut structural steel plate. The stem post may be a combination of two or more of the above fabrications, welded and scarfed together to form a rigid, central unit that maintains the continuity of keel strength up to the main deck.

An example of the bow and stem assembly structure is shown in figure 5-30, which is representative of World War II destroyer construction. Notice that the lower portion consists of a flat plate stem lying in the longitudinal, vertical centerplane and attached to the keel structure at its base. At its upper end, the structure tapers in to a rolled steel plate that forms the upper portion of the stem.

In larger vessels, in place of the vertical plate at the keel, heavy plates are used with a greater number of breast hooks to tie in to the longitudinal members and to conform to the greater dimensions of the bulb (see figure 5-31).

Regardless of the construction technique used, the bow and stern serve to extend the centerline continuity of the keel and tie the ends of the hull structure together in a faired, streamlined form.

BOTTOM AND SIDE PLATING

The bottom shell on most ships is formed by the outer and inner bottom plating, or tank tops. The inner bottom plating performs another function by providing the

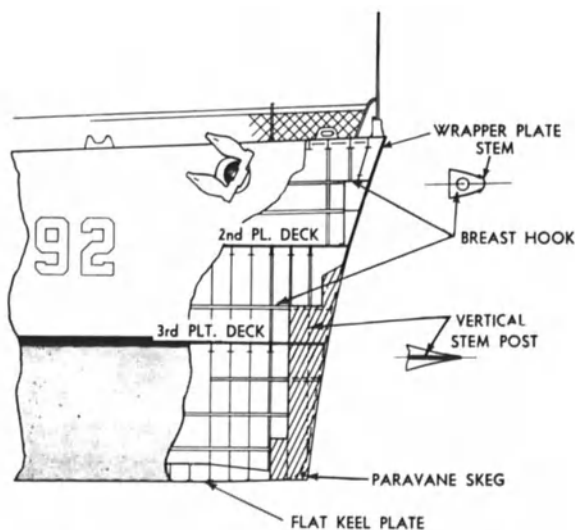


Figure 5-30. Bow and stem construction of a destroyer

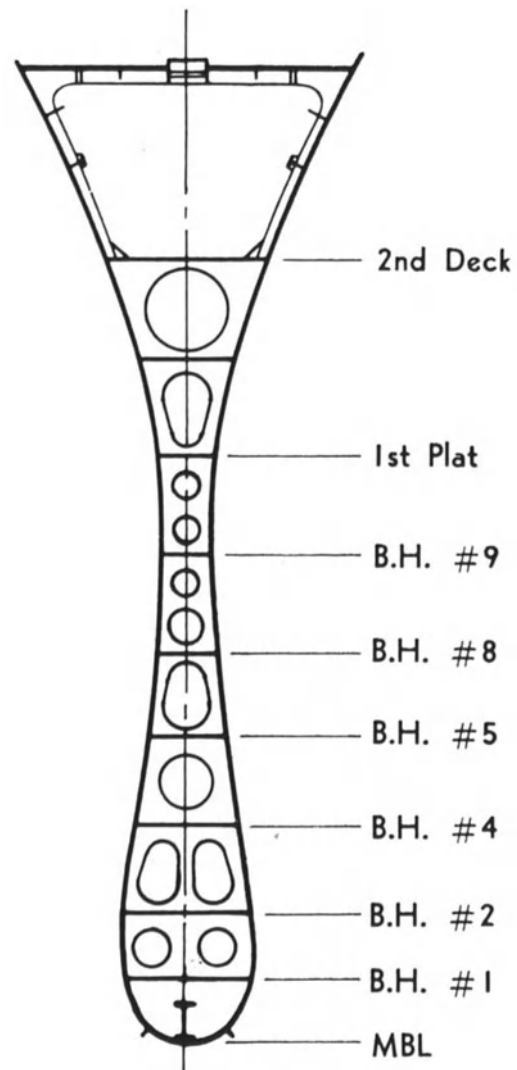


Figure 5-31. Forward frame structure of a large ship

ship with an inner watertight shell, adding to underwater protection. In the largest combatant ships, the bottom is sometimes subdivided into a double layer of bottom tanks by three layers of plating, the innermost of which is called the third bottom, or *triple bottom*.

The shell plating on both the bottom and sides is worked in longitudinal rows called *strakes*, similar to the fore and aft planking on old wooden vessels. Until recently it was common practice to rivet the seams of one or more strakes of shell and strength deck plating, and in some cases to have the bilge strake serve as crack arrestors. Current crack-arrestor practice is to weld in a strake made of HY-80 steel or other special steels. There are various plating systems in use, which are readily recognizable by inspection. Figure 5-32 indicates the arrangement of the rows of strakes. The number of strakes depends, of course, on the size of the ship. Figure 5-32 is typical of a destroyer type. Figure 5-33 indicates the heavier

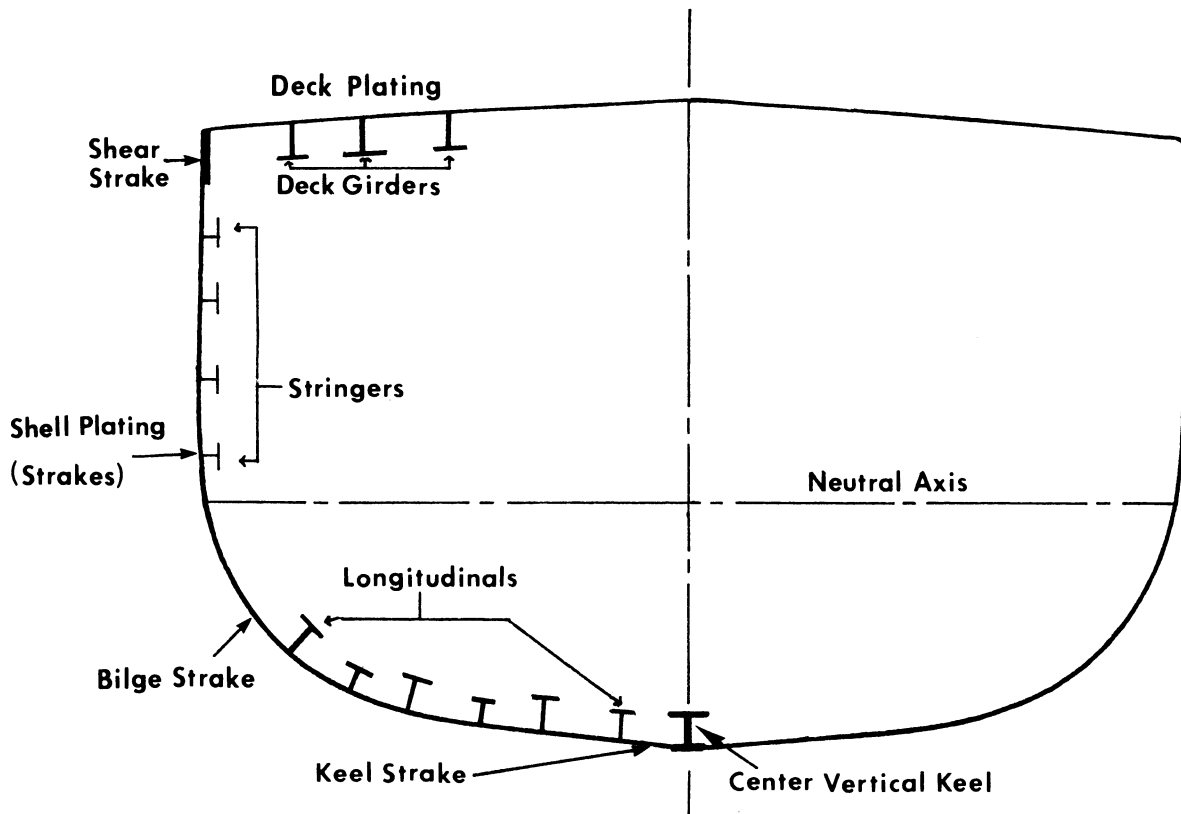


Figure 5-32. Shell section of a destroyer

structure of a larger ship. In any event, the ship of the individual plates that make up the strakes is roughly rectangular, but near the ends of the ship, where the greatest curvature is found, these plates will have to be tapered and formed, either hot or cold, in rolls and presses to conform to the ship's curvature. In any given strake, adjoining plates are attached to each other in a connection called a *butt*. These butts are generally welded flush in all vessels where fuel economy and/or high speed are important. Where two adjoining plates in a strake are lapped, as in older practice, the projecting plate edges below the waterline and transverse to the flow of water are chamfered off to an angle of 45° .

Where penetrations through the shell plating are required, as with sea chests, evaporator blow downs, sanitary drains, etc., the plating must be reinforced. When the opening is greater than 24 inches in a transverse direction, insert or doubler plates (making up for the area lost in the opening) and internal framing around the opening must be used. For smaller openings, as in the case of sea chests, for example, the sea-chest fitting itself may in many cases be considered sufficient reinforcing material.

In general, the requirements regarding the locations of the butts and seams of the shell plating with respect to the adjacent structural frame are such that they avoid

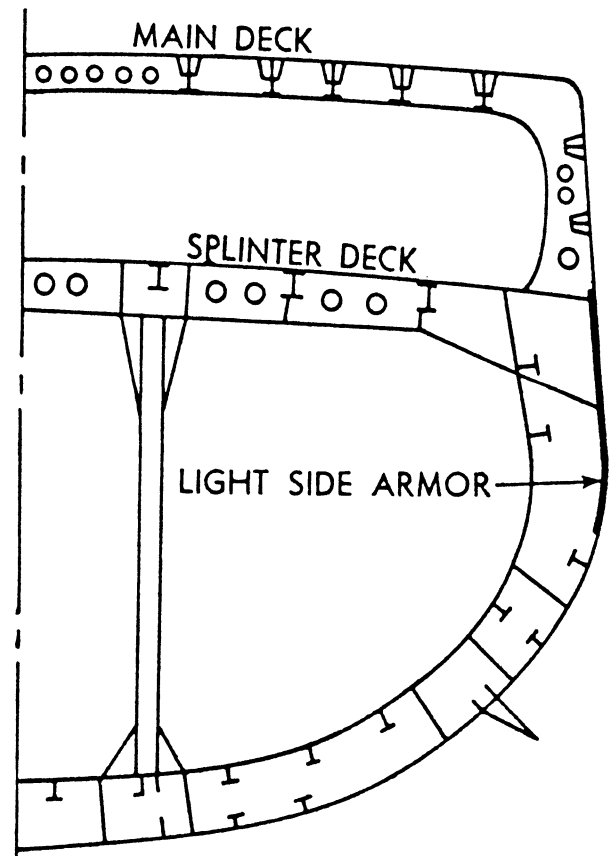


Figure 5-33. Shell section showing deep frames, inner bottom, and heavy deck beams

interference with longitudinals, bulkheads, decks, and other structural members.

The plating is thickest amidships, where the bending moment is also greatest. This thickness is usually maintained for the middle 40 percent of the ship's length and is then tapered down in gradual increments forward and aft to preserve the structural continuity of the ship's girder.

DECK PLATING

The strength deck, forming the top flange of the ship's girder, is supported by deck beams and deck girders, the former usually being tied in with the frame heads by brackets. The deck plating is laid on in a way similar to that of the side plating in fore and aft rows.

The lower decks, whose primary purpose is to divide the ship horizontally into various levels for work or other activities, are similar to but usually lighter in weight than the main or strength deck. The plating is similarly carried on deck beams and fore and aft girders. The deck beams are usually connected to the frames with welded brackets. In all strength-deck plating, the outboard strake adjacent to the shell is of heavier plating than the others and is known as the *deck stringer*.

The *sheer strake* in the side shell and the main-deck stringer plate along the deck edge form the upper corner of the ship's box girder. Because this location is farthest from the ship's neutral axis, the structure of the ship where the sides and main strength deck meet is of ex-

treme importance in the maintenance of longitudinal strength continuity.

In naval vessels, it is a rigid design and building practice not to allow a penetration or an abrupt change in dimension to limit the continuity of the deck stringer plate. Any opening in the deck interrupts the continuity of the structural members of the deck. The openings caused by hatches, boiler uptakes, barbettes, etc. may cut through several deck beams and deck girders. Consequently, the material surrounding the opening must be reinforced to supplant the material eliminated, so that the section in the vicinity of the opening will not be weakened and the stress distribution will be as uniform as possible.

Along the sides of the openings are fitted short deck girders, called *carlings* or *headers*, running at right angles to the deck beams and welded to them. Figure 5-34 shows the cellular structure formed by the deck beams and girders. Sometimes intermediate deck beams are fitted where the opening does not extend fully between the primary deck beams. The plating about the opening is increased in thickness to compensate for the discontinuity of the regular plating.

THE STRENGTH DECK AND SUPERSTRUCTURE

The uppermost continuous deck, referred to as the *strength deck*, serves in most ships to complete the enclosure of the box girder and the continuity of the ship's structure. In most naval combat vessels the superstruc-

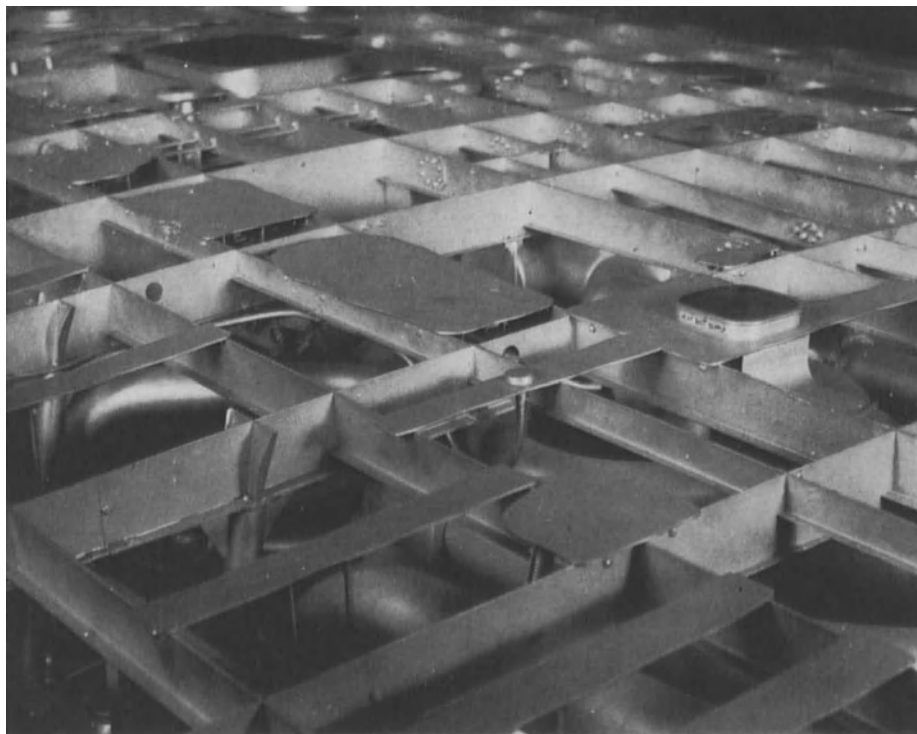


Figure 5-34. Deck structure looking forward from port quarter, showing cellular construction formed by deck beams and girders

ture, deck houses, gun-mount enclosures, etc., do not extend from rail to rail nor are they continuous throughout the length of the ship. They are, for the most part, nonwatertight spaces designed to house and shelter various ships' activities, and are usually assumed not to contribute to the structural strength of the ship. Consequently, they are made of light steel plate or aluminum, except where armor is necessary. Where the length of deck houses or similar superstructure is considerable, expansion joints are sometimes provided so that hull-bending stresses are not induced throughout the superstructure.

If the superstructure comprises less than 10 percent of the ship's length, as do the deckhouses of many cargo ships and the island structure of large aircraft carriers, the hull-bending strains transferred to a steel superstructure can be accommodated without expansion joints. An alternative to expansion joints for long superstructures is to utilize high-strength aluminum alloy materials that experience about one-third the stress levels that steel would experience if subjected to the same strain caused by hull deflections. If such aluminum alloys are used for a comparatively low-profile superstructure that extends to the full beam of the ship, the structural continuity can be carried into the uppermost deck. This concept was used in the construction of the SS *United States* in the early 1950s. This form of superstructure must be designed to withstand the stresses induced at increased distances from the neutral axis. The primary advantage of including the superstructure in the strength structure of the hull is that of greater overall strength with little increase in structural material. The theory is simply to increase the vertical dimension or depth of the girder. The extensive use of aluminum instead of steel above the main deck in ships contributes substantially to improving stability and reducing weight. The disadvantages are that aluminum has a low melting point and loses its strength more readily than steel in a fire, along with other thermal and corrosion problems mentioned in chapter 4.

Generally speaking, the strength deck in any ship should be as high as possible and yet compatible with stability considerations. Prior to the 1950s, U.S. aircraft carriers were constructed with the main or hanger deck as the strength deck. The post-World War II aircraft carriers now use the flight deck as the strength deck. Structural material high in the ship will have a pronounced effect on the center of gravity. Consequently, like most other phases of the ship's design, it is a well-considered compromise.

BULKHEADS AND VERTICAL SUPPORT

Longitudinal and transverse bulkheads provide four important functions within the ship. First, they act as stiff-

eners for the entire hull structure; second, they distribute the vertical forces of weight and buoyancy through the hull's structure; third, they serve to separate the ship's activities and functions; and finally, they provide watertight subdivision. Some bulkheads, such as the main transverse bulkheads, are constructed to withstand the hydrostatic pressure and fluctuating stresses that might be imposed through flooding in free communication with the sea.

The first function (stiffening the hull's structure) is accomplished by the *main transverse bulkheads* that extend continuously through the watertight volume of the ship from keel to main deck. Figure 5-35 shows a main transverse bulkhead at the forward end of the machinery spaces in a destroyer. These bulkheads resist the torsional and racking stresses in the structure, since they are stiffened both vertically and horizontally by stiffeners supported at the bottom, sides, and deck by brackets. (Torsion refers to twisting about an axis, and racking refers to loads that tend to distort a rectangular section into a parallelogram.) The main transverse bulkheads are joined to all the structural members, plating, and longitudinal bulkheads they contact. Where horizontal access or openings are required in bulkheads, the same means of preserving strength continuity in the decks is used.

Longitudinal bulkheads are generally not continuous in the vertical direction unless they are side bulkheads providing space for liquid stowage and voids in the side protective layer discussed in chapter 10. In such case, they will extend continuously from the bottom to the second deck or just above the waterline.

Bulkheads that are a structural part of the ship's girder are called *structural* or *strength bulkheads*. Those that act strictly as partitions or divisions are called *non-structural bulkheads*. There are also special bulkheads on some combat vessels that are a part of the torpedo protective system, called *protective* or *splinter bulkheads*. *Swash bulkheads* are used to reduce sloshing in tanks with a free surface, and *screen bulkheads* are used to subdivide cargo in bulk ships.

Nonstructural bulkheads are usually fabricated of light plate, often of stainless steel, aluminum, corrugated steel, or wire mesh. Usually, only light stiffeners are provided to prevent distortion. They are limited to the height of one deck and are not generally watertight.

Deck beams are supported at the sides by the frames to which they are attached. Between bulkheads, deck girders are supported vertically by *pillars* or *stanchions*. Stanchions are columns fitted in a vertical line from the highest deck down to the floors or frames to aid in carrying the vertical load. It is important that between-deck pillars be located vertically in line over lower pillars to insure continuous support from the longitudinals and floors up to the highest supported deck.

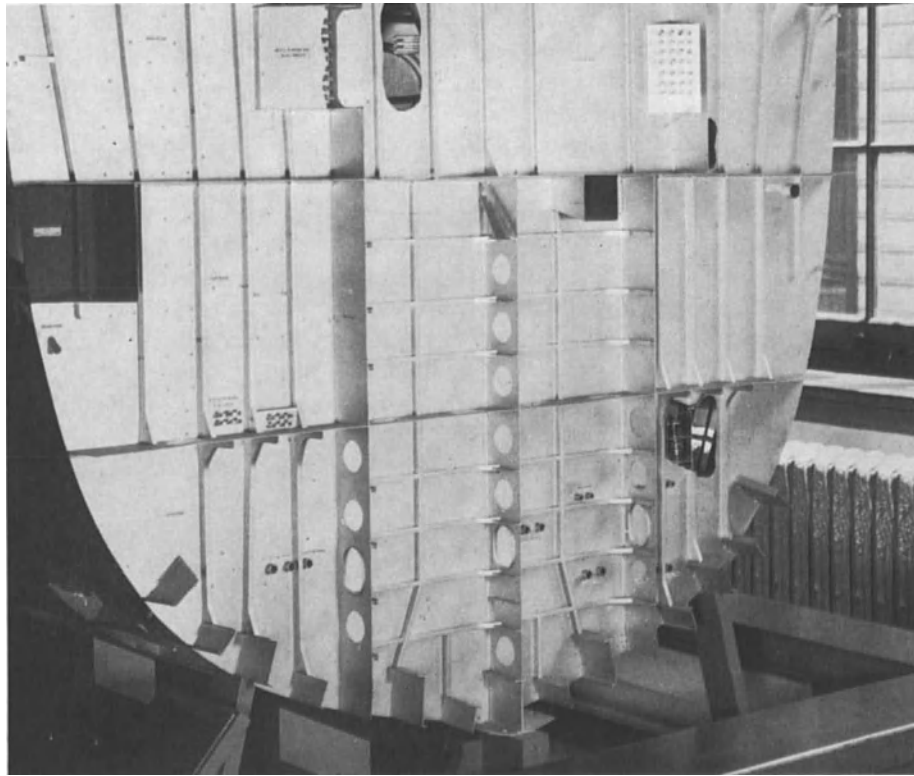


Figure 5-35. Main transverse bulkhead. (Note vertical and horizontal bulkhead stiffeners.)

5-10 Basic Structure of Naval Submarines

The structure of a naval submarine consists of a watertight envelope, which is designed to resist the operational hydrostatic pressure down to the *collapse depth* of the submarine. The principal elements are stiffened cylindrical sections, stiffened conical or tapered sections, hemispherical or elipsoidal closed-end sections, and stiffened external pressure tanks with wing (flat-plate) bulkheads. Additionally, there is a secondary structure, which does not withstand the submerged sea pressure, called the *nonpressure hull* or *outer hull*.

The primary structural components, as in any ship, are the hull plating, hull stiffeners, and bulkheads. In

the case of submarines, however, the thickness of the pressure hull plating is considerably greater than the hull plating for a surface ship, since it is designed to resist the hydrostatic loads of the design depths. This heavy steel shell, well over one inch thick, is further strengthened by circular ring frames positioned externally and/or internally, depending on location (see figure 5-36). These hull frames are of either T- or H-cross section and are either rolled or welded-up shapes. They are generally spaced 0.1 diameters apart. Further wing bulkheads are placed to form tank boundaries, and additional stiffening with the shaped-end closure bulkheads completes the watertight hull.

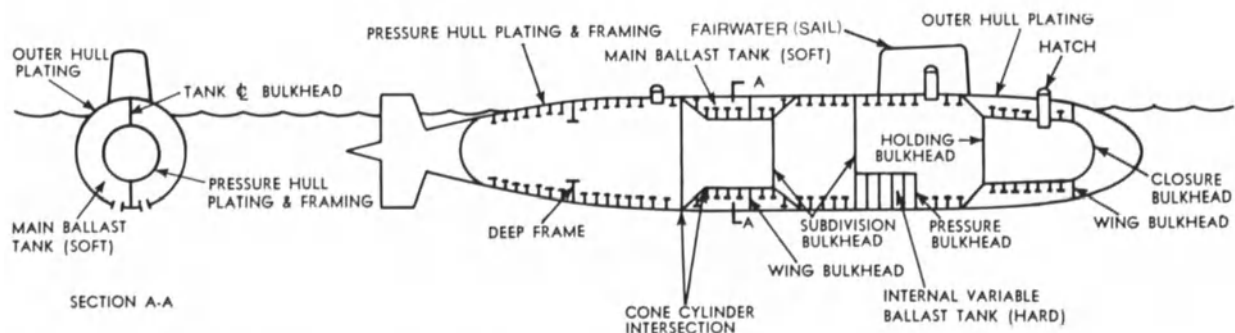


Figure 5-36. Structural profile of a submarine

PRESSURE HULL

The pressure hull (inner hull) must be strong enough to withstand hydrostatic pressures of deep submergence (test depth pressure) (see figure 5-36). The principal structures associated with the pressure hull include the transverse bulkheads, which subdivide the submarine's length into watertight compartments, and circular transverse frames, both inside and out, which strengthen the hull plating and prevent collapse when subject to test depth pressures. The pressure hull must form a watertight shell completely enclosing the operating spaces of the ship.

OUTER HULL

The outer hull forms the external boundary of the submarine, except for appendages such as the superstructure and fairwater (sail) (see figure 5-36).

The principal outer hull structure includes a system of frames and bulkheads to subdivide the enclosed volume into ballast tanks. Certain portions of the outer hull external to the pressure hull are also subject to test depth pressures. These tanks (hard tanks) have heavily constructed frames and bulkheads that are almost a continuation of pressure hull framing.

APPENDAGES

Appendages are the fairwater control surfaces (rudder and planes), piping, and other gear external to the outer hull.

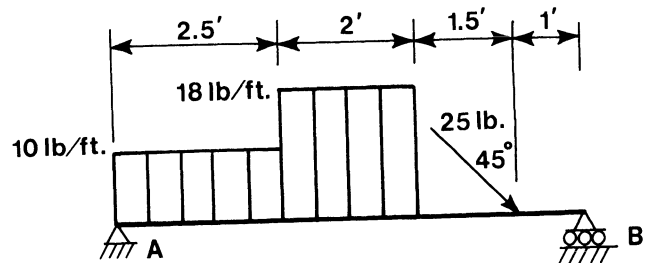
The superstructure and fairwater are constructed of lightweight plating, since they are not subjected to any severe stresses and are not an integral part of the vessel's strength members. The superstructure and fairwater are streamlined for minimum resistance.

TANKAGE

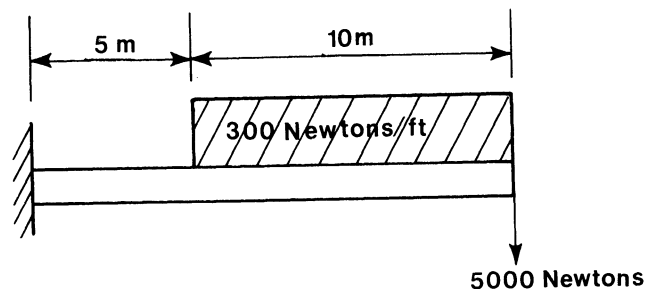
Generally speaking, submarine tankage can be separated into two classes: high-pressure tanks and non-pressure tanks. High-pressure tanks are heavily constructed tanks built to withstand test depth pressures. During normal submerged operations these tanks may be full, partially full, or empty. Nonpressure tanks are of light construction and, though exposed to the sea, are not subjected to hydrostatic pressures. During normal submerged operations, these tanks are always completely filled with seawater. On diesel submarines, non-pressure tanks not directly connected to the sea, such as normal fuel oil tanks, are equipped with seawater compensating lines to admit seawater and maintain pressure equilibrium. The function of the tanks is discussed in chapter 9.

Problems

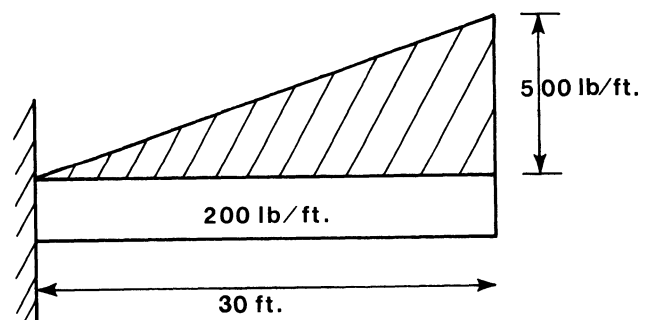
5-1. Determine the reactions at *A* and *B* for the simply supported beam in figure 5-a. Consider the beam to be weightless.



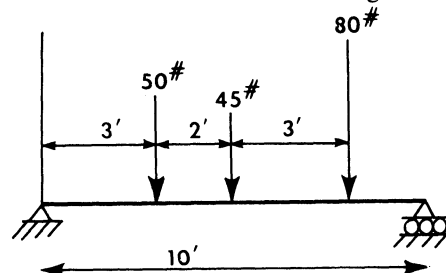
5-2. Determine the fixed end reaction in the weightless cantilevered beam in figure 5-b.



5-3. Determine the fixed end reaction in the cantilevered beam shown in figure 5-c.

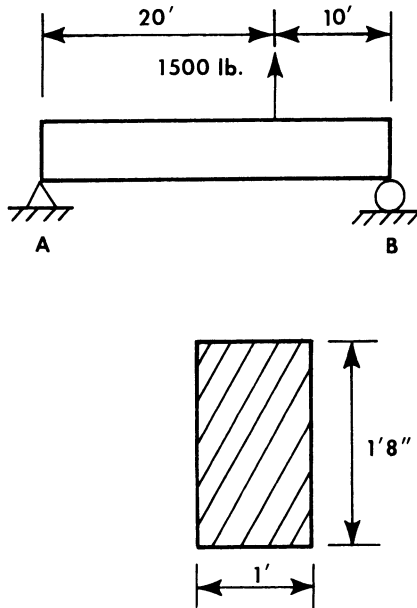


5-4. A weightless, simply supported beam with three external concentrated loads is shown in figure 5-d. Con-

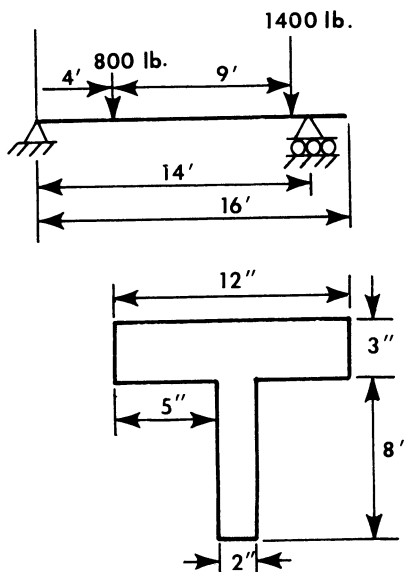


struct the load, shear, and bending-moment diagrams and indicate the maximum values for shear and bending moment.

5-5. A simply supported beam that weighs 200 lb/ft is acted upon by a concentrated upward load of 1,500 pounds, as shown in figure 5-e. The cross section of the rectangular beam is given as 1 foot wide and 1 foot 8 inches high. Determine the shear and bending-moment diagrams for the loaded beam and the maximum bending stress, assuming that the elastic flexure formula is valid for this loading.



5-6. A simply supported T-beam that weighs 73 lb/ft is acted upon by concentrated loads as shown in figure 5-f. Draw the load, shear, and bending-moment diagrams for the loaded beam. Determine the maximum bending stress in the beam, assuming that the elastic flexure formula is valid for this loading. The neutral axis of the beam is located 7.81 inches above the base



and the moment of inertia of the beam about the neutral axis is 447.4 in⁴.

5-7. Two 500-ton (long ton) liquefied natural gas (LNG) containers (figure 5-g) are to be shipped on a rectangular barge 250 feet long, 70 feet wide, and 10 feet high from Baltimore to Norfolk for installation in an LNG carrier. The barge draws 4 feet in seawater when unloaded. The containers can be considered to uniformly load the compartment in which they are located. Make the appropriate calculations and plot the load, shear, and moment diagrams, designating values of the shear and moment at each 50-ft station and at amidships.

5-8. A 200-ft-long by 35-ft-wide by 12-ft-high barge is loaded as shown in figure 5-h. The barge draws 3 feet in seawater when unloaded and 9 feet when loaded. Make the appropriate calculations and plot the load, shear, and moment diagrams, designating values of the shear and moment at each 40-ft station and at amidships.

5-9. A 1,500-long-ton nuclear reactor and 1,000 tons of associated equipment is to be shipped as a unit on a barge 250 feet long, 70 feet wide, and 12 feet high from Baltimore to the lower part of the Chesapeake Bay for further work. The barge shown in figure 5-i draws 3 feet in seawater when unloaded. It consists of five compartments, each 50 feet long and 70 feet wide. The reactor vessel can be considered to be uniformly distributed over the middle 50-ft compartment and the two 500-ton loads are distributed uniformly over the adjacent 50-ft compartments. Make the appropriate calculations and plot the load, shear, and bending-moment diagrams, designating values of the shear and moment at each 50-ft station and at amidships.

5-10. A seagoing oil barge 350 feet long by 50 feet wide and 15 feet high is loaded as shown in figure 5-j. The barge draws 3.5 feet when unloaded (empty) and currently displaces 5,250 tons in the loaded condition. Make the appropriate calculations and plot the load, shear, and moment diagrams, designating values of the shear and moment at each 70-ft station and at amidships.

5-11. A box-shaped barge is 100 feet long, 35 feet wide, and 10 feet deep. It is loaded as shown in figure 5-k and floating in salt water with $T_m = 5$ feet. Draw the load, shear, and bending-moment curves. What is the maximum bending stress if the neutral axis is 4 feet above the keel and for the cross section $I = 5.0 \times 10^5$ in⁴?

5-12. A box-shaped barge with $B = 35$ feet is loaded as shown in figure 5-l. It is floating in salt water with $T_m = 4$ feet. Draw the load, shear, and bending-moment curves. What is the maximum bending stress if the neutral axis is 3 feet above the keel and for the cross section $I = 10 \times 10^4$ in⁴?

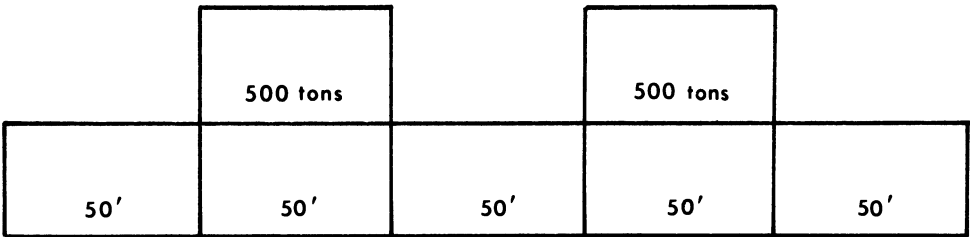


Figure 5-g.

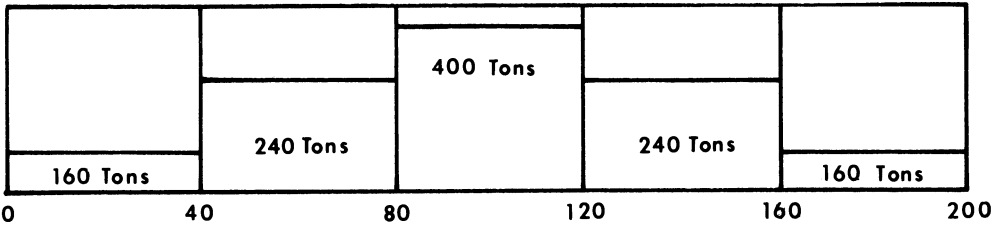


Figure 5-h.

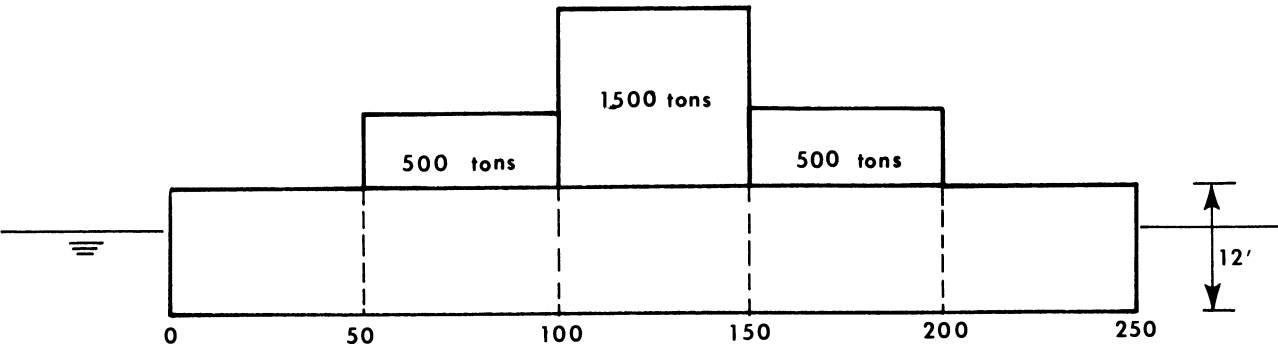


Figure 5-i.

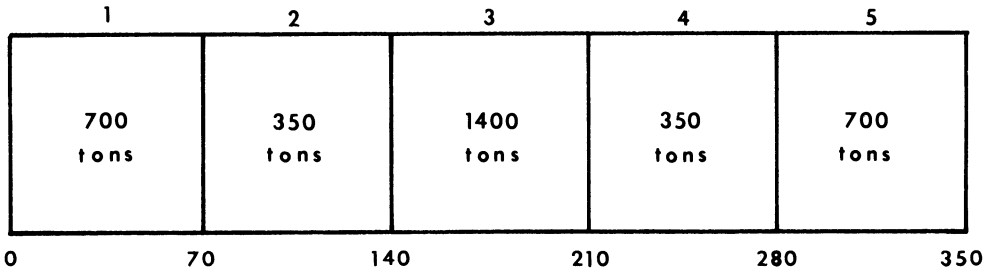


Figure 5-j.

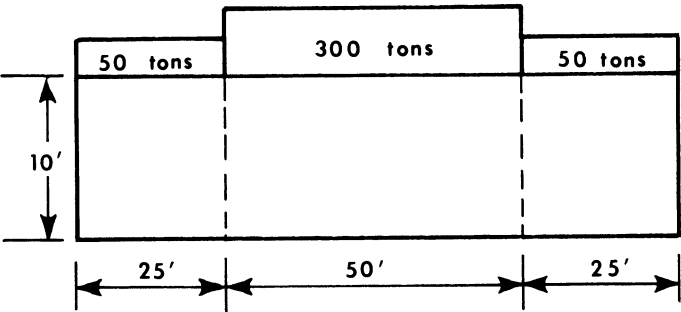


Figure 5-k.

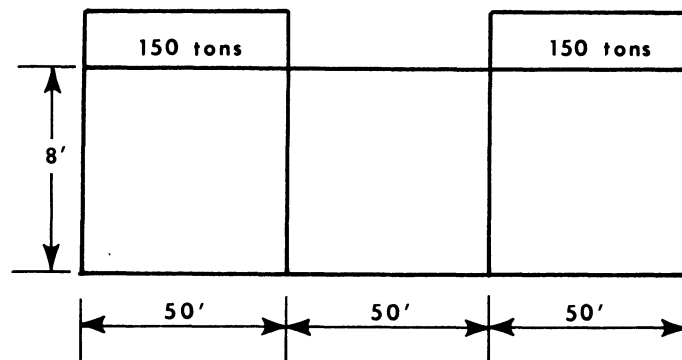


Figure 5-1.

Ship Hydrostatics and Initial Stability

6-1 Equilibrium and Stability

In chapter 2 the condition for static equilibrium was defined in terms of the balance of forces and moments. From Newton's laws of motion, it is seen that a body can be at rest or moving at constant speed only if the sum of all forces and moments acting on the body is equal to zero.

The concept of stability is somewhat more complex. Here, one is concerned with whether or not a body will return to an initial state of static equilibrium when disturbed by an unbalanced force or moment. While in the broader sense equilibrium refers to an overall balance of forces, which involves no acceleration or deceleration, *static equilibrium* is defined as follows: *A body at rest is said to be in static equilibrium.*

If this body is disturbed by an outside force and returns to its original position when the force is removed, it is said to be in *stable equilibrium*. An example of this condition is a round ball lying in an upward facing bowl, as in figure 6-1(a). The ball will always return to its rest position when disturbed by an outside force. Figure 6-1(b) illustrates the condition of *neutral equilibrium*. The ball lying on a flat horizontal plane will come to rest at any point on the plane if motion is started and then stopped by an outside force (including friction). *Unstable equilibrium* is illustrated in figure 6-1(c), where a round ball is balanced on top of an inverted bowl. Any slight disturbance of the balanced position will result in the ball rolling off the bowl.

For floating objects, the condition of stable equilibrium is illustrated by all vessels that tend to return to the original upright position when inclined by an external force. Neutral equilibrium is exhibited by a cylindrical, homogeneous cylinder floating in water that will come to rest at any position if imparted motion is stopped. Unstable equilibrium frequently occurs when a child attempts to place a doll in a toy boat before

floating it in a bathtub. When released, the toy boat inclines until it capsizes or loses its occupant, at which point the boat may return to an upright position.

THE BASIS FOR SHIP EQUILIBRIUM

Consider a ship floating upright on the surface of motionless water. In order to be at rest or in equilibrium, there must be no unbalanced forces or moments acting on it. There are two forces that maintain this equilibrium: the force of gravity and the force of buoyancy. When the ship is at rest, these two forces are acting in the same vertical line, and in order for the ship to float in equilibrium, they must be exactly equal numerically as well as opposite in direction.

The *force of gravity* acts at a point or *center of gravity* where all of the weights of the ship may be said to be concentrated. Gravity always acts vertically downward.

The *force of buoyancy* acts through the center of buoyancy, where the resultant of all of the buoyant forces is considered to be acting. This force always acts vertically upward. When the ship is heeled, the shape of the underwater body is changed, thus moving the position of the center of buoyancy.

Now, when the ship is heeled by an external inclining force and the center of buoyancy has been moved from the centerline plane of the ship, there will usually be a separation between the lines of action of the force of gravity and the force of buoyancy. This separation of lines of action of the two equal forces, which act in opposite directions, forms a couple whose magnitude is equal to the product of one of these forces (that is, displacement) and the distance separating them. In figure 6-2(a), where this moment tends to restore the ship to the upright position, the moment is called a *positive righting moment*, and the perpendicular distance between the two lines of action is the *righting arm* ($\overline{GZ}$).

Suppose now that the center of gravity is moved

upward to such a position that when the ship is heeled slightly, the buoyant force acts in a line through the center of gravity. In the new position, there are no unbalanced forces, or in other words, the ship has a zero moment arm and a zero moment. In figure 6-2(b) the ship is in neutral equilibrium, with both the righting moment and the righting arm equal to zero.

If one moves the center of gravity still higher, as in figure 6-2(c), the separation between the lines of action of the two forces as the ship is inclined slightly is in the opposite direction from that of figure 6-2(a). In this case, the moment does not act in the direction that will restore the ship to the upright, but rather will cause it to incline further. In such a situation, the ship has a negative righting moment, or *capsizing moment*, and a *negative righting arm* ($\overline{GZ}$).

These three cases illustrate the forces and relative position of their lines of action in the three fundamental states of equilibrium.

THE POSITION OF THE METACENTER AND EQUILIBRIUM

The metacenter M , discussed in chapter 3, is defined as the intersection of the vertical through the center of buoyancy of an inclined body or ship with the upright vertical when the angle of inclination approaches zero as a limit. This intersection then lies on both the line of action of the center of gravity when the ship is upright and the line of action of the buoyant force.

Consequently, it can be readily seen from the previous section that when the metacenter is above the center of gravity, as in figure 6-2(a), there is a positive righting moment formed when the ship is inclined, and the ship is in stable equilibrium.

When the metacenter and the center of gravity coincide, as in figure 6-2(b), no moment is produced and the ship is in neutral equilibrium.

When the metacenter is below the center of gravity, as in figure 6-2(c), a negative or capsizing moment is formed, and the ship is in unstable equilibrium.

In considering this relation between the metacenter and the ship's state of equilibrium, it is necessary to remember that the definition of the metacenter is actually valid only for angles of inclination from 0° up to the range of 7° to 10° . Beyond this, the intersection of the lines of action of the center of buoyancy and the vertical centerplane of the ship has no significance. Therefore, the use of the relative positions of the metacenter and the center of gravity as a criterion of stability is limited to small angles of inclination. Obviously stability itself cannot be limited to such a restricted range. Consequently, one must differentiate between overall stability at any angle of inclination and initial stability at small angles of inclination ($\phi < 10^\circ$).

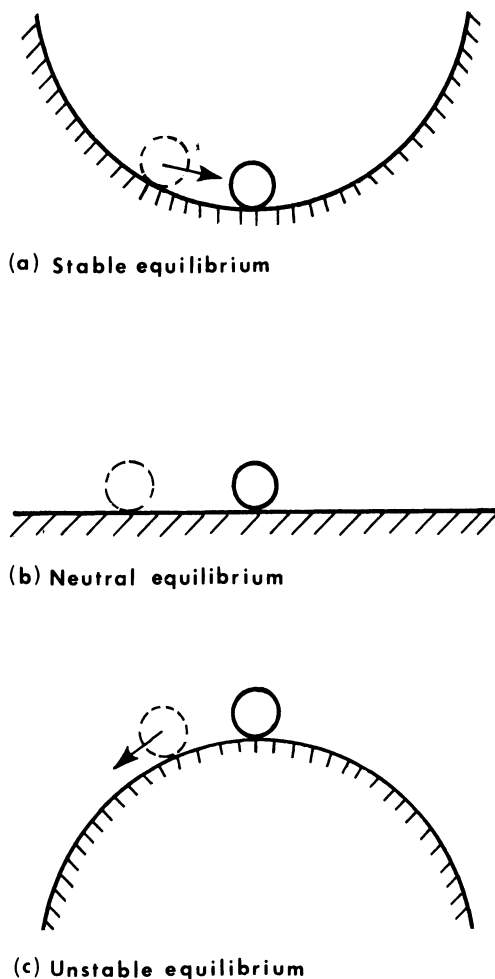


Figure 6-1. Static equilibrium: stable equilibrium (a), neutral equilibrium (b), unstable equilibrium (c)

6-2 Metacentric Height: A Measure of Initial Stability

The *metacentric height*, both transverse and longitudinal, is defined as the distance between the center of gravity and the transverse or longitudinal metacenter, measured vertically in the upright equilibrium position.

In figure 6-3, the metacentric height is $\overline{GM}$, with the ship's center of gravity at either G or G_1 . Unless otherwise specified, the metacenter and metacentric height refer to the transverse metacentric height. If the longitudinal metacenter is being discussed, the associated metacentric height is designated $\overline{GM}_L$ and spoken of as the longitudinal metacentric height.

If M is above G , the metacentric height is positive. If M is below G , $\overline{GM}$ is negative.

$\overline{GM}$ is the measure of the initial stability or the ability of the ship to resist initial heel from the upright position. A ship with a positive $\overline{GM}$ will tend to float upright and will resist initial inclining forces. A ship with a negative $\overline{GM}$ will not float upright and may be said to be initially unstable. Some ships develop a negative $\overline{GM}$ because of a condition of off-center loading and become

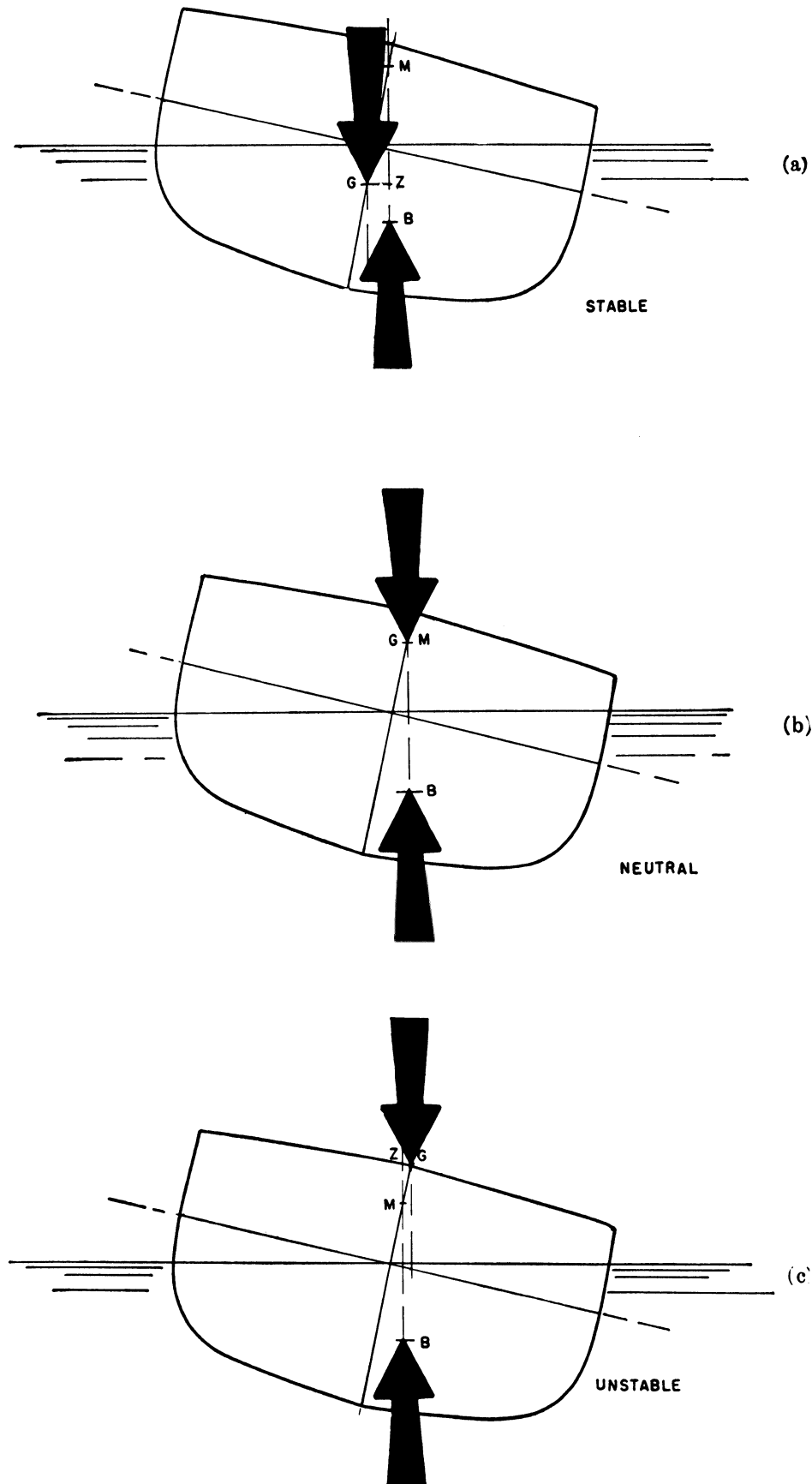


Figure 6-2. Stable (a), neutral (b), and unstable (c) equilibrium in the upright position. The hull is shown inclined by an outside force to demonstrate the tendency in each case.

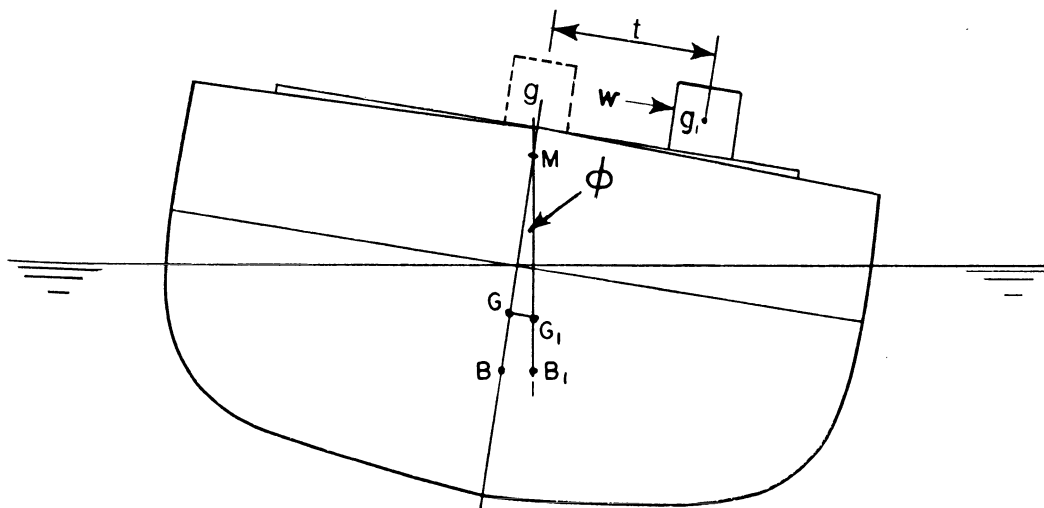


Figure 6-3. Inclined equilibrium

unstable in the upright position. Because of the change in the underwater hull form with angle of inclination, such a ship will list to either port or starboard until it reaches a point of stable equilibrium.

Since the longitudinal metacenter M_L is always located quite high above the ship (figure 6-4), it is possible to state that a negative longitudinal metacentric height $\overline{GM}_L$ will not occur under normal conditions. Longitudinal stability is discussed in the next chapter.

6-3 Righting Arm

The couple formed by the forces of buoyancy and gravity in the above discussion is, quantitatively, the product of the weight of the ship and the distance between the two forces. The perpendicular distance between the lines of action of the two forces is commonly called the righting arm ($\overline{GZ}$). When the weight or displacement of the ship is constant, we can use the value of $\overline{GZ}$ as a measure of the static stability through all angles of inclination.

For small angles of inclination (that is, where the line of action of the buoyant force when inclined intersects the vertical centerline at M),

$$\overline{GZ} = \overline{GM} \sin \phi \quad (6-1)$$

And the righting moment = $\Delta \overline{GZ}$

$$RM = \Delta \overline{GM} \sin \phi \quad (6-2)$$

where ϕ is the transverse angle of inclination in degrees (figure 6-5).

Therefore, the value of $\overline{GM}$ may be used in comparing the initial stability of ships of same type and size.

RANGE OF STABILITY

Range of stability is defined as that *range of inclination in degrees either to port or starboard, from the position of equilibrium, through which the ship is statically stable* (figure 6-6).

The magnitude of this range is primarily dependent upon the beam, freeboard, watertight integrity of the

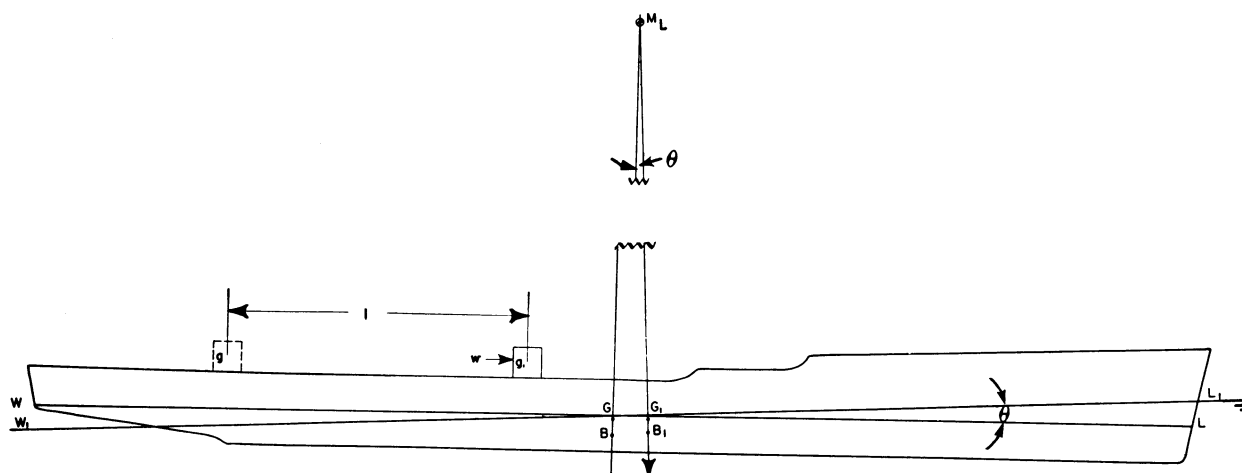


Figure 6-4. Longitudinal stability

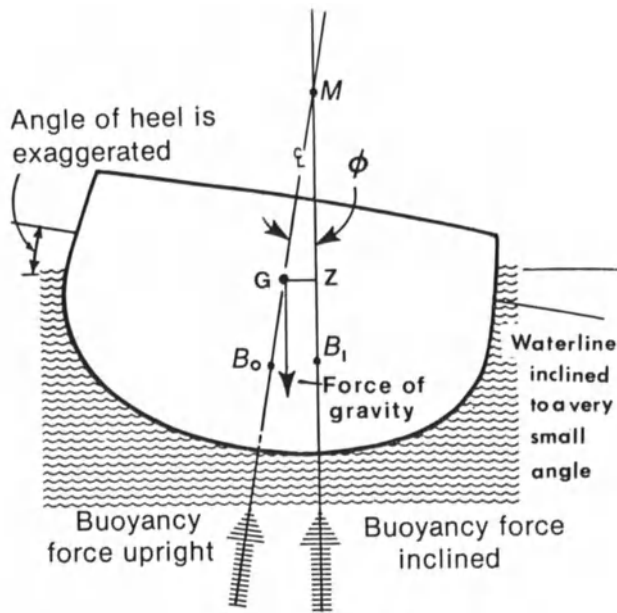


Figure 6-5. Righting arm at small angles of heel

deck and superstructure, and location of the center of gravity. This range shows the extent of positive righting arm and does not mean safety of heel throughout without capsizing. Theoretically, the range of stability indicates an angle to which the ship may be gradually inclined—without capsizing—in calm, motionless water by inclining moments not exceeding the righting moment at any angle. The righting-arm curve also shows the angle at which the righting arm is maximum. The range of stability, the maximum righting arm, the angle at which it occurs, and the area under the curve are all crucial elements in assessing the intact stability of a ship. This will be discussed in chapter 8.

6-4 Initial Stability: The Computation of the Metacentric Radius and the Location of M

Initial stability is discussed before overall stability, because it logically follows the discussion of equilibrium. Metacentric height, the measure of initial stability, is an important tool in buoyancy and stability calculations, but the overall stability (to be discussed subsequently) is the complete measure of the ability of the ship to resist inclining moments.

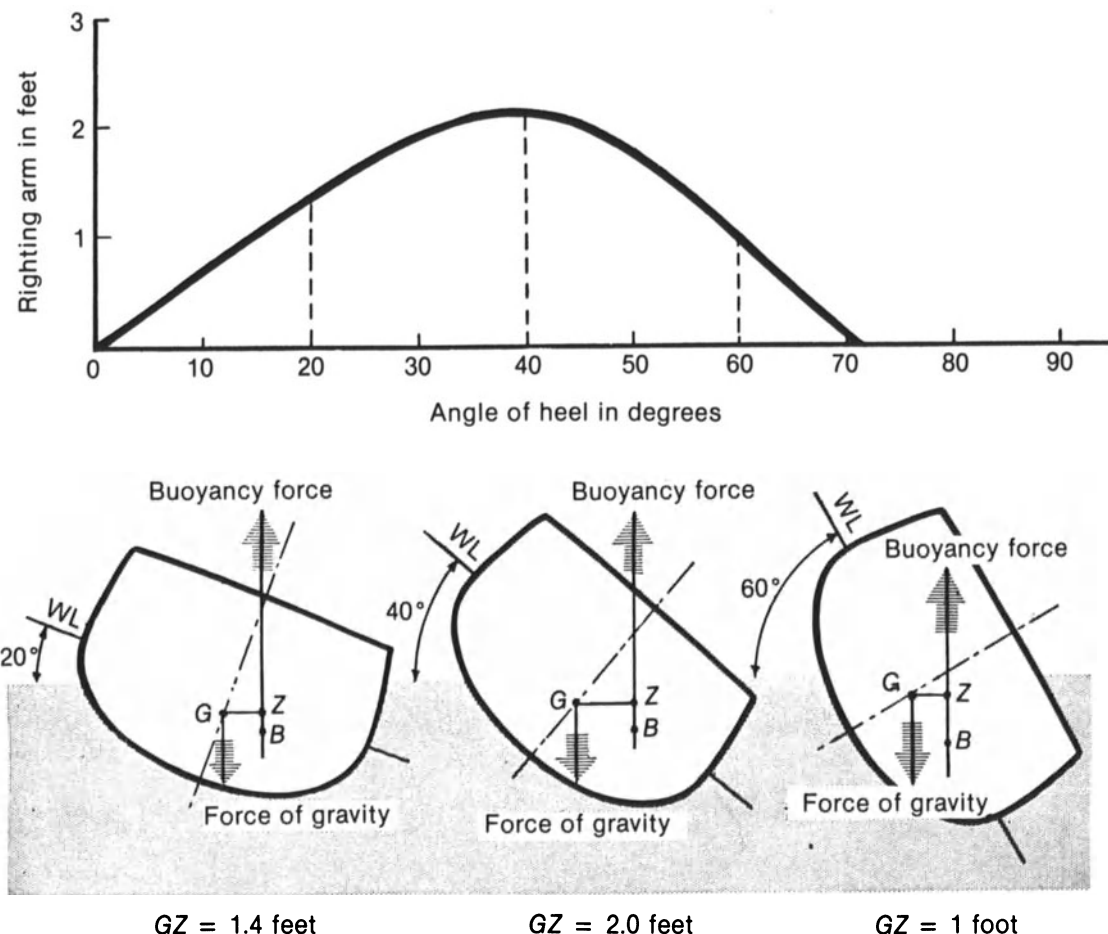


Figure 6-6. Range of stability: righting arms of a ship inclined at successively larger angles of heel

In order to numerically fix the value for metacentric height, either transverse or longitudinal, one must actually locate the metacenter and the center of gravity in relation to some fixed datum plane, preferably a horizontal plane through the bottom of the flat keel amidships. These values are normally called $\overline{KM}$, $\overline{KM}_L$, and $\overline{KG}$, respectively. The methods of computing these values will be given in the discussions immediately following.

Figure 6-7 shows diagrammatically the cross section of a ship and her waterplane shape. The ship is heeled to a small angle of inclination by an external moment so that she floats at the waterline W_1L_1 instead of at WL . The location of the center of buoyancy shifts from B to B_1 as previously discussed.

- Let r = half-breadth of the ship
 $\delta\phi$ = small angle of inclination in radians
 n = center of gravity (area) of triangle LOL_1
 dx = increment of length L
 ∇ = volume of displacement
 ρ = density of water.

Now, the area of the triangle LOL_1 is approximately $\frac{1}{2}r r\delta\phi$. The distance from the apex O to the center of gravity n equals $\frac{2}{3}r$ for a small angle $\delta\phi$. The moment of area of triangle LOL_1 about the longitudinal centerline plane is

$$\left(\frac{1}{2} r r\delta\phi\right) \overline{On}$$

$$\left(\frac{1}{2} r r\delta\phi\right) \frac{2}{3} r$$

The moment of volume of the wedge (having triangle LOL_1 as its section and dx as its thickness) about the longitudinal centerline plane is

$$\left(\frac{1}{2} r r\delta\phi\right) \left(\frac{2}{3} r\right) dx$$

or for the entire ship, we have by integration

$$\int_0^L \frac{1}{2} r r\delta\phi \frac{2}{3} r dx$$

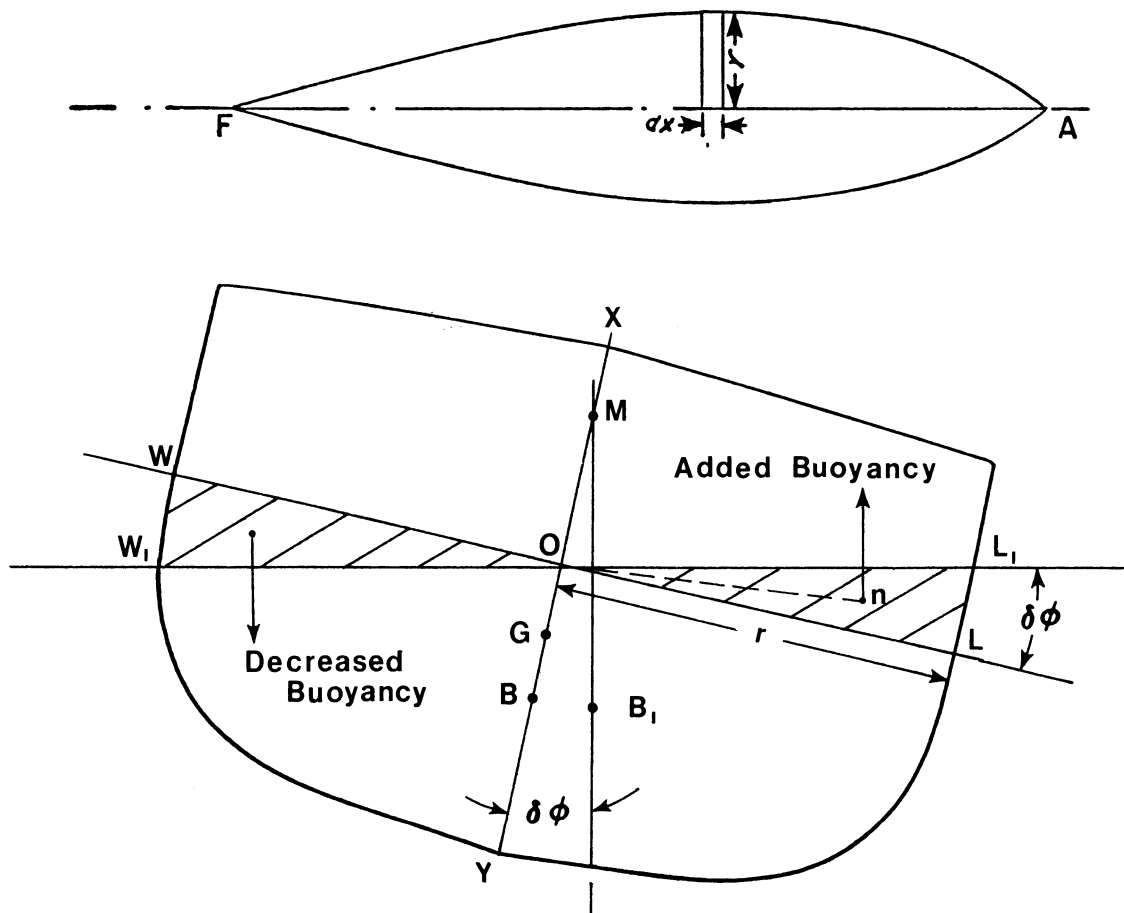


Figure 6-7. Computation of metacentric radius $\overline{BM}$

Because the volume of the emerged wedge WOW_1 is equal to the immersed wedge LOL_1 , the added buoyancy due to LOL_1 is exactly equal to the lost buoyancy due to WOW_1 . Hence, there are two equal moments acting in the same direction about the centerline plane.

Therefore, the total moment is

$$2 \int_0^L \frac{1}{2} r r \delta \phi \cdot \frac{2}{3} r \cdot dx$$

or,

$$\int_0^L \frac{2}{3} r^3 \delta \phi dx$$

This total moment, or moment of both wedges, is what causes the center of buoyancy to move from B to B_1 . The moment of the underwater volume with new center at B_1 taken about the original center at B is $\nabla \cdot \overline{BB_1}$. This must equal the moment of both wedges. Therefore,

$$\nabla \overline{BB_1} = \int_0^L \frac{2}{3} r^3 \delta \phi dx$$

By geometry,

$$\overline{BB_1} = \overline{BM} \sin \delta \phi$$

For small angles, $\sin \delta \phi \approx \delta \phi$

Therefore,

$$\overline{BB_1} = \overline{BM} \delta \phi$$

and

$$\begin{aligned} \nabla \overline{BB_1} &= \nabla \overline{BM} \delta \phi = \delta \phi \int_0^L \frac{2}{3} r^3 dx \\ \overline{BM} &= \frac{\int_0^L \frac{2}{3} r^3 dx}{\nabla} \end{aligned}$$

Since $\int_0^L \frac{2}{3} r^3 dx$ is the expression for moment of inertia of the waterline plane about the longitudinal centerline I (equation 2-34), then

$$\overline{BM} = \frac{I}{\nabla} \quad (6-3)$$

Similarly, it may be shown that

$$\overline{BM}_L = \frac{I_L}{\nabla} \quad (6-4)$$

where $\overline{BM}_L$ = the longitudinal metacentric radius
 I_L = the moment inertia of the waterplane about a transverse axis through the center of flotation.

For the purpose of estimating initial stability during the early design phases of a new ship, the block coefficient

$$C_B = \nabla / LBT$$

and the waterplane inertia coefficients

$$C_{IT} = 12 I_T / B^3 L$$

and

$$C_{IL} = 12 I_L / BL^3$$

(equations 3-5a and b) may be substituted in equations 6-3 and 6-4 to yield

$$\overline{BM} = \frac{C_{IT} B^3 L / 12}{C_B L B T} = \frac{C_{IT} B^2}{C_B 12 T} \quad (6-5a)$$

$$\overline{BM}_L = \frac{C_{IL} B L^3 / 12}{C_B L B T} = \frac{C_{IL} L^2}{C_B 12 T} \quad (6-5b)$$

To get a parametric estimate of $\overline{KM}$, equation 6-5 can be combined with equations 3-11 and 3-10a.

$$\begin{aligned} \overline{KM} &= \overline{KB} + \overline{BM} \\ \overline{KM} &= \frac{C_{WP} T}{C_{WP} + C_B} + \frac{C_{IT} B^2}{C_B 12 T} \quad (6-6) \end{aligned}$$

Note, however, that C_{WP} , C_B , and C_{IT} are themselves functions of drafts, except for special cases of rectangular barges and wall-sided ships (figure 3-14). For illustrative purposes, a rectangular barge or *box-shaped lighter (BSL)*, with

$$C_B = C_{WP} = 1.0$$

and

$$C_{IT} = C_{IL} = 1.0$$

is used to simplify calculations, since for this shape,

$$\overline{KB} = T/2$$

For this shape only,

$$\overline{KM} = \overline{KB} + \overline{BM} = T/2 + B^2/12T \quad (6-7)$$

$$\overline{KM}_L = \overline{KB} + \overline{BM}_L = T/2 + L^2/12T \quad (6-8)$$

Example 6-1: Determine the initial transverse stability of a 922-ton ship in salt water that is 240 feet long, has a beam of 28 feet, and a draft of 8 feet. The center of buoyancy is 4.2 feet above the keel and KG is equal to 8.0 feet. The offsets from the 8-ft waterline with a station spacing of 20 feet are given in table 6-1, along with the approximate integration of the moment of inertia, using the trapezoidal rule from appendix B.

Table 6-1

Half Ordinate	Cube of Half Ordinate	Multiplier	Function of Cubes $f(I)$
.5	0.125	$\frac{1}{2}$	0
2.5	15.6	1	16
6.6	287.5	1	288
10.1	1,030	1	1,030
12.5	1,953	1	1,953
13.7	2,571	1	2,571
14.0	2,744	1	2,744
13.9	2,686	1	2,686
13.0	2,197	1	2,197
10.8	1,260	1	1,260
6.6	287.5	1	288
2.1	9.26	1	9
0.5	0.125	$\frac{1}{2}$	0

Solution:

$$\Sigma f(I) = 15,042$$

Thus,

$$I = \int_0^L \frac{2}{3} r^3 dx = \frac{2}{3} h \quad f(I) = \frac{2}{3} \cdot 20 \cdot 15,042$$

$$= 200,560 \text{ ft}^4$$

Since $\Delta = 922$ tons,

$$\nabla = 922(35 \text{ ft}^3/\text{ton}) = 32,270 \text{ ft}^3$$

Thus,

$$\overline{BM} = \frac{I}{\nabla} = \frac{200,560}{32,270} = 6.22 \text{ ft}$$

$$\overline{KM} = \overline{KB} + \overline{BM} = 4.2 + 6.22 = 10.4 \text{ ft}$$

$$\overline{GM} = \overline{KM} - \overline{KG} = 10.4 - 8 = \boxed{2.4 \text{ feet}}$$

Alternatively, $\overline{BM}$ could have been calculated using equation 6-5, if C_B and C_{IT} had been given.

$$C_B = \frac{\nabla}{LBT} = \frac{32,270}{240(28)8} = .6$$

$$C_{IT} = \frac{12I_T}{B^3L} = \frac{12(200,560)}{(28)^3 240} = .457$$

and

$$\overline{BM} = \frac{C_{IT}B^2}{C_B 12T} = \frac{.457(28)^2}{.6(12)8} = 6.22 \text{ ft}$$

as before.

Example 6-2: Determine the initial stability of a homogeneous rectangular block of wood with square ends floating in fresh water as a function of the specific gravity of the wood.

Solution: The wood will float only if the specific gravity γ lies between 0 and 1.0. It will float at a draft T based on γ as illustrated in figure 6-8(a).

weight = buoyancy

$$LBH(62.4 \gamma) = LBT(62.4)$$

$$T = \gamma H$$

but

$$H = B$$

$$T = \gamma B$$

Using equation 6-7,

$$\overline{KM} = \overline{KB} + \overline{BM}$$

$$= \frac{T}{2} + \frac{B^2}{12T}$$

$$= \frac{\gamma B}{2} + \frac{B^2}{12\gamma B}$$

Since $\overline{KG} = H/2 = B/2$,

$$\overline{GM} = \overline{KM} - \overline{KG}$$

$$= \frac{\gamma B}{2} + \frac{B}{12\gamma} - \frac{B}{2}$$

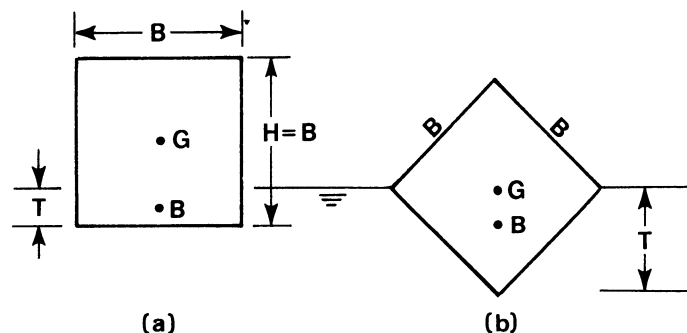


Figure 6-8. Stability of a homogeneous block of wood

$$= \frac{B}{12\gamma} (6\gamma^2 - 6\gamma + 1)$$

Thus, positive initial stability ($\overline{GM}$ greater than zero) will result when

$$(6\gamma^2 - 6\gamma + 1) > 0$$

Negative initial stability ($\overline{GM}$ less than zero) will result when

$$(6\gamma^2 - 6\gamma + 1) < 0$$

and the block will be neutrally stable when

$$6\gamma^2 - 6\gamma + 1 = 0$$

or

$$\gamma = \frac{6 \pm \sqrt{6^2 - 4(6)}}{2(6)}$$

$$\gamma = \frac{1}{2} \pm \frac{\sqrt{3}}{6} = \boxed{0.21 \text{ or } 0.79}$$

To check these results, consider a large rectangular block, 24 inches on each side with a $\overline{KG}$ equal to 1.0 feet. If the log is of balsa wood,

$$\gamma = 0.13$$

$$T = 0.13(2) = 0.26 \text{ ft}$$

$$\overline{GM}_1 = \overline{KB}_1 + \overline{BM}_1 - \overline{KG}_1$$

$$= \frac{T}{2} + \frac{B^2}{12T} - \frac{B}{2}$$

$$= \frac{0.26}{2} + \frac{2^2}{12(.26)} - 1$$

$$= 0.13 + 1.28 - 1$$

$$= \boxed{0.41 \text{ feet}} \text{ (stable equilibrium)}$$

Thus, the log will float as illustrated in figure 6-8(a). If the log is made of yellow pine,

$$\gamma = 0.5$$

and set as in figure 6-8(a),

$$T = 0.5(2) = 1.0 \text{ ft}$$

$$\overline{GM}_2 = \overline{KB}_2 + \overline{BM}_2 - \overline{KG}_2$$

$$= \frac{0.5}{2} + \frac{2^2}{12(.5)} - 1$$

$$= 0.25 + .67 - 1$$

$$= \boxed{-0.08 \text{ feet}} \text{ (unstable equilibrium)}$$

The log will roll over and assume the position illustrated in figure 6-8(b). Note that the new metacentric radius will be increased by the effect of the larger beam even though the draft increased. The new beam equals $\sqrt{2} B$.

$$T_3 = \frac{\sqrt{2}}{2} B$$

$$\overline{KB}_3 = \frac{2}{3} T = \frac{\sqrt{2}}{3} B$$

$$\overline{BM}_3 = \frac{I}{\nabla} = \frac{(\sqrt{2}B)^3 L / 12}{LB^2 / 2}$$

$$= \frac{\sqrt{2}}{3} B$$

$$\overline{KG}_3 = \frac{\sqrt{2}}{2} B$$

$$\therefore \overline{GM}_3 = \overline{KB}_3 + \overline{BM}_3 - \overline{KG}_3$$

$$= \frac{\sqrt{2}}{3} B + \frac{\sqrt{2}}{3} B - \frac{\sqrt{2}}{2} B$$

$$= \frac{\sqrt{2}}{6} B$$

$$= \boxed{0.47 \text{ feet}} \text{ (stable equilibrium)}$$

with the pointed end down.

Thus,

$$\overline{GM} > 0 \text{ for } \gamma < 0.21 \text{ and } \gamma > .79$$

and

$$\overline{GM} < 0 \text{ for } 0.21 < \gamma < .79$$

6-5 Initial Stability: The Effect of Changes in the Center of Gravity

The center of gravity is the center through which all the weights constituting the ship and its contents may be assumed to act. It is generally calculated by summing moments about three mutually perpendicular axes if the location of each and every weight in the ship is known. The location of the vertical center of gravity $\overline{KG}$ is obtained by summing moments about the keel using the equations introduced in chapter 2 (figure 6-9).

$$\overline{KG} = \frac{\sum w_i \overline{Kg}_i}{\sum w_i} \quad (6-9)$$

where $\Delta = \sum w_i$

In the case where only two weights, w_1 and w_2 , and two locations are involved, equation 6-9 reduces to

$$\overline{KG} = \frac{w_1 \overline{Kg}_1 + w_2 \overline{Kg}_2}{w_1 + w_2}$$

This approach may be used to determine the new $\overline{KG}$

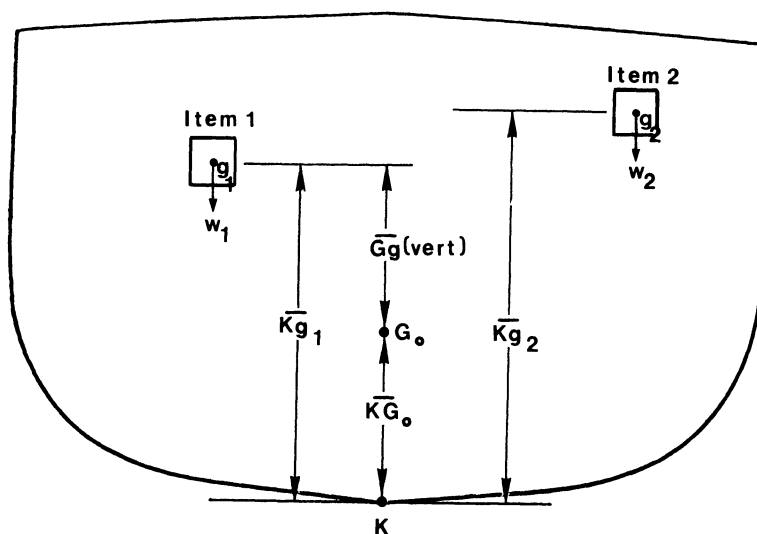


Figure 6-9. Center-of-gravity calculations

caused by a weight addition or removal from a ship when the original $\overline{KG}_o$ and Δ_o are known. Let

$$w_2 = \Delta_o$$

and

$$\overline{Kg}_2 = \overline{KG}_o$$

Then

$$\overline{KG}_1 = \frac{\Delta_o \overline{KG}_o + w_1 \overline{Kg}_1}{\Delta_1} \quad (6-10)$$

where $\Delta_1 = \Delta_o + w_1$

If a weight removal is involved, the sign before w_1 should be changed to a minus sign. Multiple weight additions and removals may be handled by combining equations 6-9 and 6-10, using summations with the proper sign for additions or removals.

Weight additions may also be analyzed by taking moments about the center of gravity of the original condition. This utilizes the physics principle that *the shift in position of the center of gravity of a system of weights caused by the addition of a single weight is equal to the moment of the added weight about the center of gravity of the original system divided by the total weight of the final system*. In the vertical direction,

$$\overline{G_o G_1}(\text{vert}) = \frac{w \overline{Kg}_1}{\Delta_o + w} \quad (6-11)$$

Thus,

$$\overline{KG}_1 = \overline{KG}_o + \overline{G_o G_1}(\text{vert}) \quad (6-12)$$

The equivalence of equations 6-12 and 6-10 may be demonstrated as follows by combining 6-11 and 6-12:

$$\overline{KG}_1 = \overline{KG}_o + \frac{w \overline{Kg}_1}{\Delta_o + w}$$

but

$$\overline{Gg}_1(\text{vert}) = \overline{Kg}_1 - \overline{KG}_o \quad (\text{figure 6-9}) \quad (6-13)$$

$$\begin{aligned} \overline{KG}_1 &= \overline{KG}_o + \frac{w(\overline{Kg}_1 - \overline{KG}_o)}{\Delta_o + w} \\ &= \frac{(\Delta_o + w)\overline{KG}_o + w(\overline{Kg}_1 - \overline{KG}_o)}{\Delta_o + w} \\ &= \frac{\Delta_o \overline{KG}_o + w \overline{Kg}_1}{\Delta_o + w} \end{aligned}$$

as before.

Equation 6-10 has several distinct advantages over equation 6-11, namely, (1) it gives the actual final position of G with respect to the keel; (2) there is no confusion as to the direction G moves, up or down; and (3) it is particularly adaptable in grounding and dry-docking problems.

It should be noted that when a weight is added to or removed from a system of weights, the center of gravity of the system moves toward the added weight and away from the removed weight on a line between the original center of gravity and that of the added or removed weight.

The lateral position of the center of gravity may be found by summing moments about the centerline of the ship. For a ship in the upright position in stable equilibrium, the center of gravity will be located on the centerline, so the summation of moments must equal zero.

The longitudinal position of the center of gravity (LCG) may be found by summing moments about the forward or after perpendicular to avoid worrying about sign conventions, or it may be found by summing mo-

ments about the midship station $\oslash$ if the signs of the moments are carefully accounted for. The LCG, along with the LCB and LCF, are tabulated on the Maritime Administration's standard form for stability and trim calculations as distances from the forward perpendicular. Since the center of gravity must be located directly in line with the center of buoyancy when a ship is in static equilibrium, however, it is common to convert the LCG to a position relative to midships, regardless of the method of calculation, since the center of buoyancy and center of flotation are given on the curves of form relative to midships. Therefore, the standard reference axes for all stability calculations in this text will be given in a *keel-centerline-midships* axis system.

THE SHIFT OF CENTER OF GRAVITY WITHIN A SYSTEM (WEIGHT SHIFT)

Consider a system consisting of a body of weight W (figure 6-10), with the center of gravity located at G and containing a small weight w , which is available for shifting within this system originally located at g . If w is shifted from g to g_1 , a moment of magnitude $w\overline{gg}_1$ is

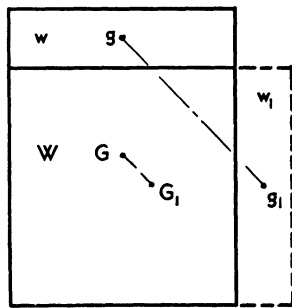


Figure 6-10. Shift of center of gravity

created, resulting in a shift of the ship's center of gravity from G to G_1 , that is, a moment of $\Delta\overline{GG}_1$. These moments must be equal.

$$\Delta\overline{GG}_1 = w\overline{gg}_1$$

or

$$\overline{GG}_1 = \frac{w\overline{gg}_1}{\Delta} \quad (6-14)$$

Note that distances $\overline{GG}_1$ and $\overline{gg}_1$ are parallel and in the same direction, regardless of the direction of shift.

To facilitate the determination of the effect of a weight shift on the draft forward and aft and on the list, it is helpful to determine the components of the shift in the ship's center of gravity parallel to the three basic planes of the ship—that is, the vertical, transverse, and longitudinal components of the shift in the center of gravity. This is accomplished by considering the weight shift to be reduced to component movements that are vertical, longitudinal, and transverse.

Consider a weight on the second deck, port side, and aft of $\oslash$ to be moved to the main deck, starboard side, and forward of $\oslash$ (figure 6-11).

First, assume the weight to be moved vertically from the second deck, point g to g_1 , at the same height that the weight will occupy on the main deck when in its final position. The ship's center of gravity will move vertically as follows:

$$\overline{GG}_1 (\text{vert}) = \frac{wv}{\Delta} \quad (6-15)$$

where v = vertical distance that the weight w is shifted
 Δ = displacement of the ship.

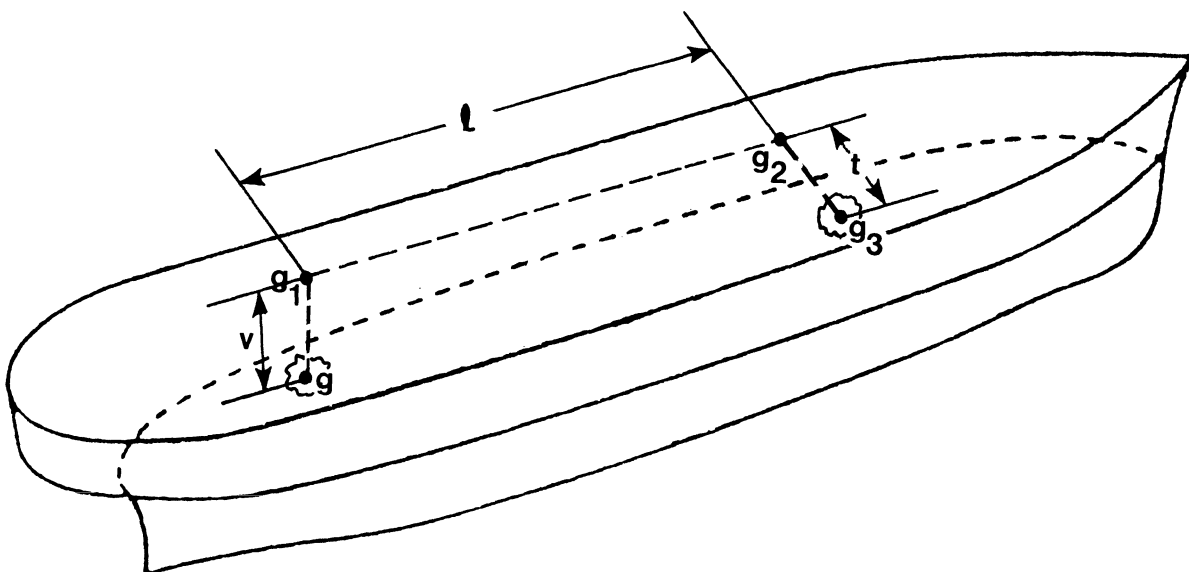


Figure 6-11. Three-dimensional weight shift

Secondly, assume the weight to be moved longitudinally from point g_1 to its final longitudinal position g_2 . The ship's center of gravity will move longitudinally as follows:

$$\overline{G_1G_2} \text{ (long)} = \frac{wl}{\Delta} \quad (6-16)$$

where l = longitudinal distance that the weight w is shifted.

Lastly, assume the weight to be moved transversely from point g_2 to its final transverse position g_3 . The ship's center of gravity will move transversely as follows:

$$\overline{G_2G_3} \text{ (trans)} = \frac{wt}{\Delta} \quad (6-17a)$$

where t = transverse distance that the weight w is shifted.

The numerical subscripts assigned to G and g have no significance other than denoting successive positions of the centers of gravity of the ship and the weight. As examples, note that in figures 6-3 and 6-4, the ship's center of gravity is denoted by G . The center of gravity moves to G_1 in each case in response to weight shifts that are longitudinal and/or transverse respectively.

6-6 Conditions of Equilibrium in an Inclined Position

A ship in equilibrium in an inclined position, as in figure 6-3, must have the center of gravity directly in a vertical line with the center of buoyancy. Referring to the diagram, if the ship is floating in equilibrium at some angle of list ϕ , the center of gravity is off the vertical centerline plane at a position G_1 .

When the ship was upright, the center of gravity was located on the vertical centerline plane at a position G .

In order to move the center of gravity from G to G_1 , a weight w within the system was shifted transversely from point g to g_1 .

Then,

$$\overline{GG_1} \text{ (trans)} = \frac{wt}{\Delta} \quad (6-17b)$$

If the shift occurs perpendicularly to the vertical centerline plane, it can be seen from figure 6-3 that

$$\overline{GG_1} \text{ (trans)} = \overline{GM} \tan \phi \quad (6-17c)$$

Then,

$$wt = \Delta \overline{GM} \tan \phi \quad (6-18)$$

where ϕ = angle of inclination (less than 7° to 10°).

For longitudinal inclinations, the same principles are applicable. Using the longitudinal metacenter M_L and the equation,

$$\overline{G_1G_2} \text{ (long)} = \frac{wl}{\Delta}$$

the following relation can be derived from figure 6-4:

$$wl = \Delta \overline{GM_L} \tan \theta \quad (6-19)$$

where θ = angle of longitudinal inclination.

A useful relation in this case is

$$\tan \theta = \frac{\text{change in trim as measured at draft marks}}{\text{length between draft marks}} \quad (6-20)$$

LIST CAUSED BY OFF-CENTER WEIGHT ADDITIONS

One method to determine list caused by off-center weight additions is to assume that the weight is added at the

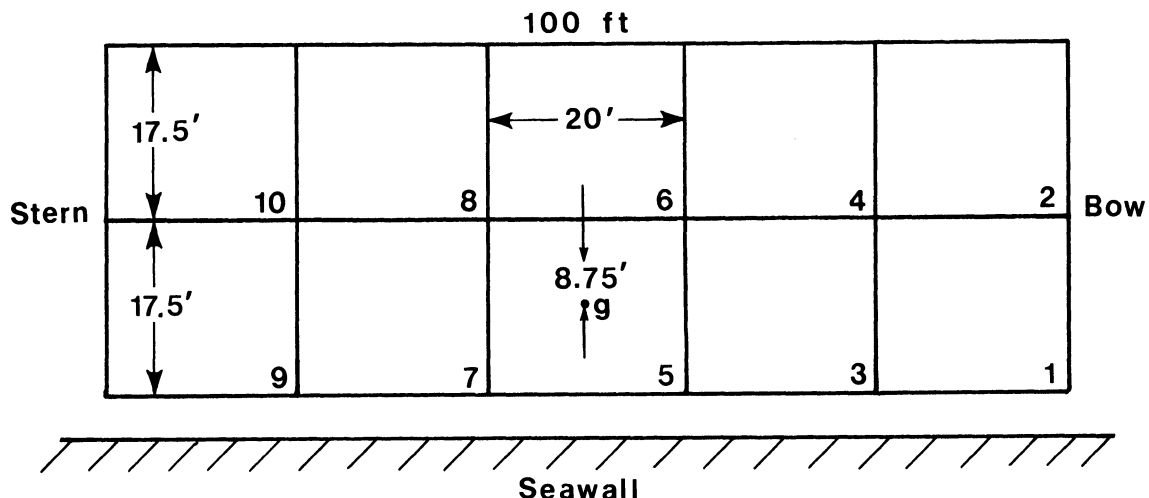


Figure 6-12. Example 6-3: oil barge (from above)

centerline of the ship, then to calculate the new drafts and new $\overline{GM}$, and move the weight to its final off-center location. The resulting list can be calculated using equations 6-17b, 6-17c, or 6-18.

Example 6-3: A box-shaped lighter (C_B is equal to 1.0) is used for oil delivery within an overseas naval base. The oil barge is 100 feet long, 35 feet wide, and 12 feet high. When empty, the barge draws 1.4 feet in salt water and has a $\overline{KG}$ of 3.3 feet. It is divided into ten tanks, each 20 feet long and 17.5 feet wide, with a capacity of 80 tons of oil when filled. The tanks are arranged as shown in figure 6-12. (The use of a centerline longitudinal bulkhead is not good design practice, as explained in chapters 5 and 10, but is used here and in example 6-5 to produce a relatively large free-surface effect.) The geometric center of each tank is 6.05 feet above the keel and 8.75 feet off the centerline. Determine the changes in the vertical and transverse position of the center of gravity of the barge and the angle of list, if the Oil King fills tank 5 to capacity with all the other tanks empty.

Solution: To find the change in vertical position of G , use equations 6-10 and 6-12.

$$\Delta_0 = \frac{C_B L B T}{35} = \frac{1.0(100)35(1.4)}{35}$$

$$\Delta_0 = 140 \text{ tons}$$

$$\Delta_1 = 140 + 80 = 220 \text{ tons}$$

$$\begin{aligned} \overline{KG}_1 &= \frac{\Delta_0 \overline{KG}_0 + w \overline{Kg}}{\Delta_1} \\ &= \frac{140(3.3) + 80(6.50)}{220} \end{aligned}$$

$$= 4.3 \text{ ft}$$

$$\begin{aligned} \overline{GG}_1 (\text{vert}) &= \overline{KG}_1 - \overline{KG}_0 \\ &= 4.3 - 3.3 = 1.0 \text{ feet} \end{aligned}$$

Or, using equations 6-11 and 6-13,

$$\begin{aligned} \overline{GG}_1 (\text{vert}) &= \frac{w \overline{Gg}_1}{\Delta + w} = \frac{w(\overline{Kg}_1 - \overline{KG}_0)}{\Delta_1} \\ &= \frac{80(6.05 - 3.3)}{220} \\ &= \boxed{1.0 \text{ ft}} \end{aligned}$$

The change in transverse position of G can be found by adding the weight at the centerline and moving it

transversely 8.75 feet, using equation 6-17b.

$$\begin{aligned} \overline{GG}_1 (\text{trans}) &= \frac{wt}{\Delta} \\ &= \frac{80(8.75)}{220} \\ &= \boxed{3.18 \text{ feet}} \end{aligned}$$

To find the angle of list, determine $\overline{GM}$, using equations 6-7 and 3-12, and then use equation 6-17c.

$$\begin{aligned} \overline{GM}_1 &= \overline{KM}_1 - \overline{KG}_1 \\ &= \frac{T_1}{2} + \frac{B^2}{12T_1} - \overline{KG}_1 \end{aligned}$$

$$\begin{aligned} T_1 &= \frac{\Delta_1 \times 35}{LB} = \frac{220(35)}{100(35)} \\ &= 2.2 \text{ ft} \end{aligned}$$

$$\begin{aligned} \overline{GM}_1 &= \frac{2.2}{2} + \frac{35^2}{12(2.2)} - 4.3 \\ &= 1.1 + 46.4 - 4.3 \\ &= 43.2 \text{ ft} \end{aligned}$$

$$\tan \phi = \frac{\overline{GG}_1 (\text{trans})}{\overline{GM}_1} = \frac{3.18}{43.2} = 0.074$$

$$\phi = \boxed{4.2^\circ \text{ to starboard}}$$

Changes in List: Analysis of Moments for Weight Removal

In determining list caused by weight removal, specific consideration must be given to the cause of list. When a list exists and is due solely to the weight that is to be removed, then the act of removing the weight results in an upright ship with G on the centerline. In this case, the procedure of determining change in list is a simple reversal again of the weight addition procedure.

However, in practice, weight removal does not normally involve the simple situation described above. One must expect that the ship in the original condition is either upright or has a list, and that the removal of a weight will cause a change in list, resulting in a final or residual list. This situation is often encountered in everyday problems of weight removal, such as the consumption of fuel and stores, unloading deck cargo, landing craft, boats, planes, and vehicles, or in problems of grounding and jettisoning. Such weight removal generally results in residual inclining moments, and any discussion of weight removal must be approached with these practical considerations in mind. A problem of weight removal resulting in a residual list may be solved as follows:

1. Determine the original off-center location of G .
2. Determine the transverse shift in G caused by removing the weight.
3. By combining the two values determined above, find the final off-center location of G and the resulting list.

Residual List by Moment Equation

The above analysis may be resolved into an equation for the residual angle of list caused by the weight removal. In considering the amount of residual list, it should be kept in mind that the change in list is the result of the moment of the removed weight. Taking this moment relative to the centerplane, it is wt , where w is the removed weight and t is the transverse distance to the centerline. It is actually the difference between the original and final transverse moments of the ship. These moments are, respectively, $\Delta \overline{GM} \tan \phi$ and $\Delta_1 \overline{G_1 M_1} \tan \phi_1$; therefore, we may write

$$\Delta \overline{GM} \tan \phi - \Delta_1 \overline{G_1 M_1} \tan \phi_1 = wt$$

$$\tan \phi_1 = \frac{\Delta \overline{GM} \tan \phi - wt}{\Delta_1 \overline{G_1 M_1}} \quad (6-21)$$

where Δ = original displacement of ship with weight on board
 w = weight removed from ship
 Δ_1 = final displacement with w off ship
 $\overline{GM}$ = original metacentric height with w on board
 $\overline{G_1 M_1}$ = final metacentric height with w off ship
 t = distance from w to centerline of ship
 ϕ = original angle of list
 ϕ_1 = final angle of list with w off ship.

Applications of the above equation are unlimited insofar as the location of weight or the original and final list is concerned (within the small angle range of the application of $\overline{GM}$). For example, if an off-center weight is removed when the ship is originally upright, the residual list is simply

$$\tan \phi = \frac{-wt}{\Delta_1 \overline{G_1 M_1}} \quad (6-22)$$

6-7 Docking

The conditions in cases of dry-docking basically concern weight removal. Essentially, the ship is only partially waterborne. The remaining portion of the ship's weight rests upon the bottom of the dock.

The problem may be most readily explained by treating it as a weight-removal problem. When the ship comes to rest along her keel and is no longer a freely floating body, she cannot maintain her original displacement. The reduction in displacement resulting from her di-

minishing draft is entirely accounted for by the upward force acting on the ship's bottom. The ship can be considered as if it were still a floating body of reduced displacement—the reduced displacement being due to the upward force at the bottom (which is considered a removed weight). If one assumes, as is normally the case, that this force acts at the first point of contact, the keel, the point at which an imaginary weight has been removed is established. An evaluation of this condition will account for the emerged layer and a new position of M and G . Combining the changes in G and M , a new transverse metacentric height is established, and if desired, a corrected theoretical static stability curve may be plotted. Because the point of application of the bottom force would generally move when the ship is heeled, a complete static stability curve taken from the upright characteristics would be of questionable accuracy.

The docking officer is concerned with the modified initial stability characteristics, that is, the effect of draft changes on $\overline{GM}$. This special form of $\overline{GM}$, designated here as $\overline{GM}_{\text{Dock}}$, will be useful in determining the tendency of the ship to stay upright on the keel blocks during the dry-docking operation and during refloating. (When aground, the same approach can be used to indicate any danger of rolling over, if the ship is not supported by a flat bottom. This will be discussed in chapter 10.)

For the special case of docking and grounding, several symbols will be introduced as follows:

F_R = ground reaction force at the keel which in turn equals the weight of the lost buoyant layer w , as illustrated in figure 6-13

F_B = net buoyant force = $\Delta_0 - F_R = \Delta_0 - w$

Δ_0 = displacement before docking

$\overline{KG}_0$ = location of the center of gravity before docking

Since the weight is considered to be removed at the keel, the application of equation 6-10 yields

$$\overline{KG}_{\text{Dock}} = \frac{\Delta_0 \overline{KG}_0 - F_R \overline{KG}}{\Delta_0 - F_R}$$

or since $\overline{KG} = 0$,

$$\overline{KG}_{\text{Dock}} = \frac{\Delta_0 \overline{KG}_0}{F_B} \quad (6-23)$$

$\overline{KG}_{\text{Dock}}$ may be thought of as an effective height of the center of gravity, even though the actual center of gravity of the weight distribution of the ship is unchanged.

Thus,

$$\begin{aligned} \overline{GM}_{\text{Dock}} &= \overline{KM}_{\text{Dock}} - \overline{KG}_{\text{Dock}} \\ &= \overline{KM}_{\text{Dock}} - \frac{\Delta_0 \overline{KG}_0}{F_B} \end{aligned} \quad (6-24a)$$

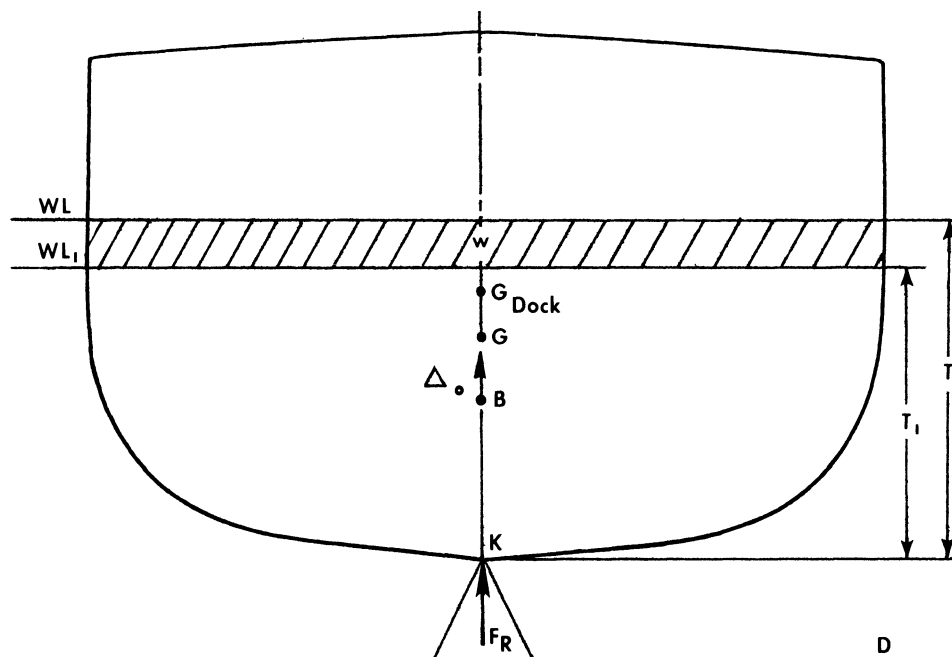


Figure 6-13. Forces in docking

where $\overline{KM}_{\text{Dock}}$ = height of metacenter above the keel at the reduced draft

or

$$\overline{GM}_{\text{Dock}} = \frac{F_B \overline{KM}_{\text{Dock}} - \Delta_0 \overline{KG}_0}{F_B} \quad (6-24b)$$

The righting moment for small angles of heel when resting on the keel blocks is

$$F_B \overline{GZ} = F_B \overline{GM}_{\text{Dock}} \sin \phi$$

or

$$F_B \overline{GZ} = (F_B \overline{KM}_{\text{Dock}} - \Delta_0 \overline{KG}_0) \sin \phi \quad (6-25)$$

The ship will become unstable if the righting arm becomes negative or if

$$\overline{GM}_{\text{Dock}} < 0$$

Thus the criteria for stability in a docking situation is

$$F_B \overline{KM}_{\text{Dock}} > \Delta_0 \overline{KG}_0 \quad (6-26)$$

The critical draft T_c for *wall-sided ships* (figure 3-14) and barges for which $C_B = C_{WP} = \text{constant}$, regardless of draft, may be found by combining equation 6-24b with equation 6-6 for

$$\overline{GM}_{\text{Dock}} = 0$$

$$F_B \overline{KM} = F_B (\overline{KB} + \overline{BM}) = \Delta_0 \overline{KG}_0$$

$$C_B \frac{LBT_c}{35} \left(\frac{C_{WP} T_c}{C_{WP} + C_B} + \frac{C_{IT} B^2}{C_B 12 T_c} \right) = C_B \frac{LBT_0}{35} \overline{KG}_0$$

$$\frac{C_{WP} T_c^2}{C_{WP} + C_B} + \frac{C_{IT} B^2 T_0}{C_B 12 T_0} = T_0 \overline{KG}_0$$

But

$$\frac{C_{IT} B^2}{C_B 12 T_0} = \overline{BM}_0$$

and

$$\frac{C_{WP}}{C_{WP} + C_B} = \frac{1}{2} \quad \text{for} \quad C_{WP} = C_B$$

$$\therefore \frac{T_c^2}{2} + T_0 \overline{BM}_0 = T_0 \overline{KG}_0$$

$$T_c = [2T_0(\overline{KG}_0 - \overline{BM}_0)]^{1/2} \quad (6-27)$$

Since many large bulk carriers and small barges approximate a wall-sided ship, this expression may be used to *estimate* the critical draft during dry-docking for flat-bottomed ships with nearly vertical sides. Note that if $\overline{KG}_0$ is less than $\overline{BM}_0$, the ship will be stable throughout, as long as it is partially waterborne.

The critical draft for ships with sufficient deadrise and hull forms that can become unstable in a docking situation can be found by using a trial-and-error iteration method or a graphical solution based on data from the curves of form. An initial estimate of the critical draft may be obtained by solving for an estimated F_B using the original $\overline{KM}_0$.

$$F_B (\text{est}) = \frac{\Delta_0 \overline{KG}_0}{\overline{KM}_0} \quad (6-28)$$

The draft that corresponds to a displacement equal to F_B can then be used as a starting point for a graphical or iterative solution, as illustrated in the following example.

Example 6-4: A DD 692 destroyer is to be dry-docked in level trim with a mean draft of 13 feet 6 inches. If its current $\overline{KG}$ equals 15.0 feet, determine the critical draft for which $\overline{GM}_{\text{Dock}}$ is equal to 0.

Solution: From the curves of form,

$$\overline{KM}_0 = 19.3 \text{ ft}$$

$$\Delta_0 = 3,180 \text{ tons}$$

Using equation 6-28 to get a critical displacement estimate,

$$\begin{aligned} F_B (\text{est}) &= \frac{\Delta_0 \overline{KG}_0}{\overline{KM}_0} \\ &= \frac{3,180(15)}{19.3} \\ &= 2,470 \text{ tons} \end{aligned}$$

which occurs near a draft of 11 feet 6 inches.

A table can now be set up to solve for the critical draft, using equation 6-26 or 6-24b as desired.

Table 6-2

T ft	F_B tons	$\overline{KM}$ ft	$F_B \overline{KM}$ ft-tons	$\Delta_0 \overline{KG}_0$ ft-tons	$\overline{GM}_{\text{Dock}}$ ft (if desired)
11.5	2,510	20.2	50,700	47,700	1.2
11.0	2,350	20.4	47,940	"	0.1
10.5	2,190	20.6	45,114	"	-1.18

The ship will become unstable at a draft of just less than 11 feet. It is sufficient simply to compare the $F_B \overline{KM}$ product with the $\Delta_0 \overline{KG}_0$ product on a calculator, but $\overline{GM}_{\text{Dock}}$ can be tabulated and a graphical solution used (figure 6-14).

The above procedure is used for estimating the critical draft at which bilge blocks and other lateral supports will begin supporting a vessel during the dry-docking operation, or the draft at which the vessel will lift off one side of the bilge blocks during refloating. However, there is much more to the dry-dock operation than this. A crucial concern of the docking officer is the current status of the $\overline{KG}$ estimate. Docking accidents shouldn't happen, but lack of communication between the docking officer and the ship superintendent caused a navy tug to capsize in a dry dock during refloating operations. The engines of the tug were removed and not replaced before the docking officer attempted to refloat the tug. Since the $\overline{KG}$ of the tug was raised by the removal of the engines, the vessel became unstable and capsized, as indicated in figure 6-15. In a sense, refloating following alterations may be more hazardous than dry-docking, if the location of the ship's center of gravity has changed to an unknown value.

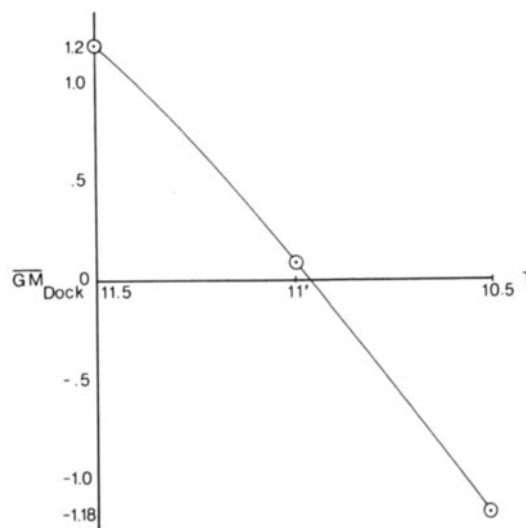


Figure 6-14. Critical draft

A more common problem in dry-docking vessels occurs when the slope of the keel blocks is not equal to the trim of the ship. Sonar domes require that the vessel be supported over less than the length of the ship, which may cause strength problems for the keel blocks near the ends of the vessel.

6-8 Free Surface

Liquid that only partially fills a compartment is said to have a free surface that will tend to remain horizontal. The effect is that when the ship is heeled, the liquid will flow to the low side (that is, in the direction of inclination) and add to the inclining moment.

Actually, the movement of the liquid is an athwartship shift of weight, which varies with the angle of inclination. However, in considering the effect of free surface, it is more convenient to consider the effect of

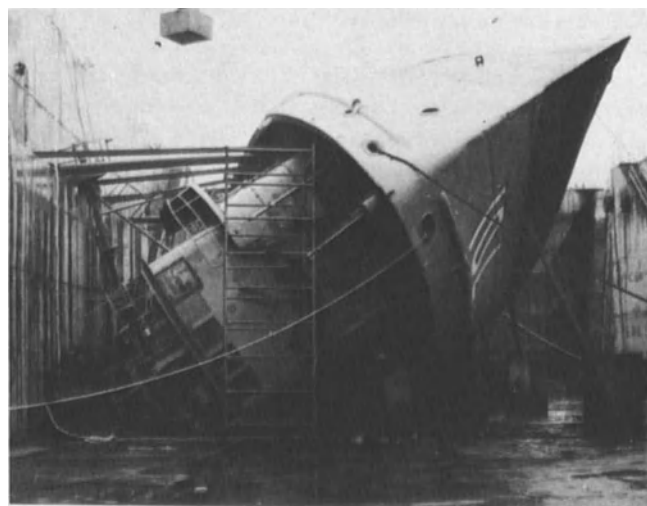


Figure 6-15. Dry-dock capsize. (Courtesy of LCDR R. C. Montgomery.)

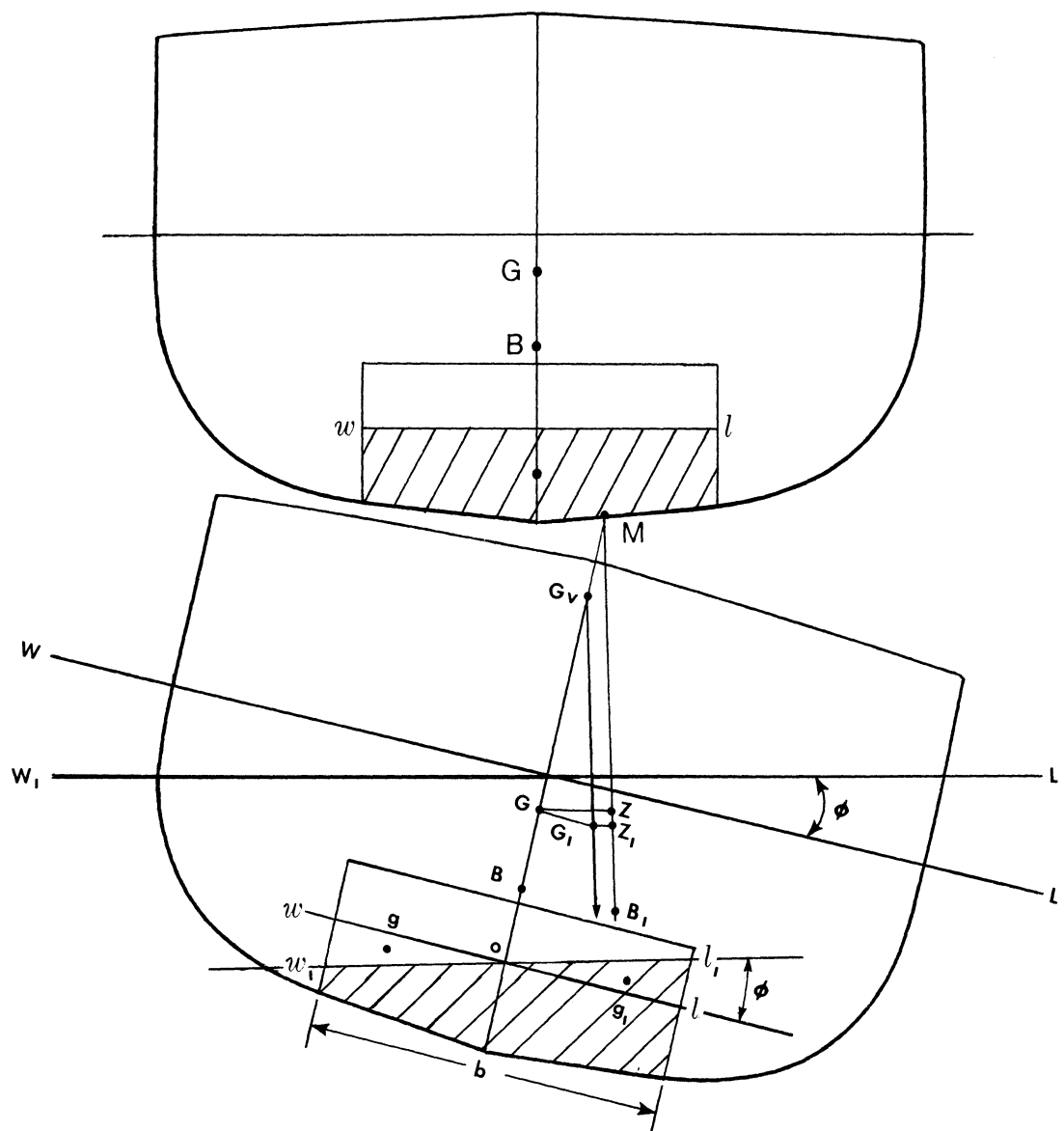


Figure 6-16. Free-surface effect

the movement of the liquid as changing the apparent height of the center of gravity.

To consider the effect on initial and overall stability, one must assume first a small angle of inclination, such that the liquid in the tank to be considered does not touch the top of the tank nor expose the bottom of the tank. Figure 6-16 shows a compartment partially filled with liquid that has a free surface wl with the ship upright. When the ship is heeled to the small angle ϕ , the free surface shifts to w_1l_1 , remaining horizontal. In referring to figure 6-16, let

- g = centroid of shifted liquid wedge wow_1 , which is "removed" as the ship heels
- g = acceleration of gravity in weight calculations
- g_1 = centroid of shifted liquid wedge lol_1 , which is "added" as the ship heels
- b = breadth of the tank at the free surface

l = length of the tank at the free surface
 i = moment of inertia of the free-surface area about the longitudinal centerline of the tank, which is equal to $\frac{b^3l}{12}$ for a rectangular tank

w = weight of liquid in the shifted wedge lol_1
 γ_t = specific gravity of the liquid in the tank
 ρ_t = density of the liquid in the tank
 γ_s = specific gravity of the ship flotation liquid
 ρ_s = density of the ship flotation liquid
 ∇_s = volume of displacement of the ship
 ϕ = angle of heel or list

$\overline{gg}_{1T}$ = transverse shift in the center of gravity of the shifted wedge relative to the original upright position

$\overline{gg}_{1V}$ = vertical shift in the center of gravity of the shifted wedge relative to the original upright position.

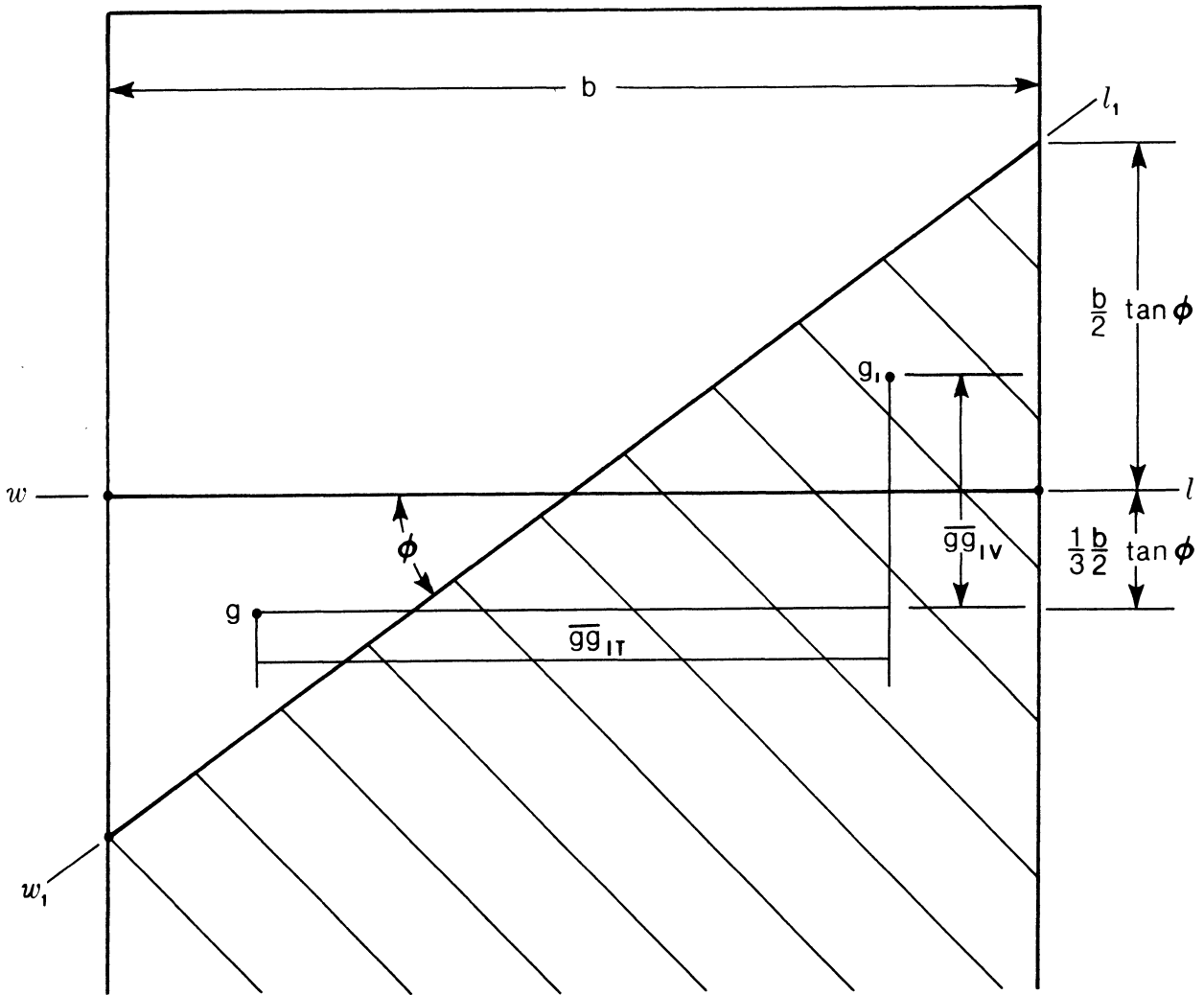


Figure 6-17. Weight shift caused by free surface in a rectangular tank

Consider a rectangular tank of width b and length l . (Other tank shapes could be analyzed, but this would require the use of integrals and the loss of simple geometric analysis.) As the ship is heeled to a small angle ϕ , the liquid in the tank must adjust so that its surface remains horizontal. This adjustment results in a transverse shift in the center of gravity of both the tank and the ship. The shift can be analyzed with reference to figure 6-17, in which the vertical scale has been exaggerated for clarity of the relationships.

$$\overline{gg}_1 \text{ (trans)} = \overline{gg}_{1T} = \frac{2b}{3}$$

$$\overline{gg}_1 \text{ (vert)} = \overline{gg}_{1V} = 2 \frac{1}{3} \frac{b}{2} \tan \phi$$

$$= \frac{2b}{3} \frac{\tan \phi}{2}$$

$$= \overline{gg}_{1T} \frac{\tan \phi}{2}$$

The weight of liquid in the shifted wedge is

$$w = \rho_l g (\text{volume}) = \rho_l g [1/2(b/2)(b/2 \tan \phi)l]$$

$$= \rho_l \frac{gb^2l}{8} \tan \phi$$

The transverse shift in the center of gravity of the ship is

$$\begin{aligned} \overline{GG}_{1T} &= \frac{w \overline{gg}_{1T}}{\Delta_1} \\ &= \frac{\left[\rho_l g \frac{b^2l}{8} \tan \phi \right] \left[\frac{2b}{3} \right]}{\rho_s g \nabla_s} \end{aligned}$$

$$= \frac{\rho_l}{\rho_s} \frac{b^3l}{12} \frac{\tan \phi}{\nabla_s}$$

$$\overline{GG}_{1T} = \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \tan \phi \quad (6-29)$$

A more general analysis for arbitrarily shaped tanks would also yield equation 6-29, with the moment of inertia of the liquid surface identified as the generalized integral in the analysis.

The vertical shift in the center of gravity of the ship is

$$\begin{aligned}\overline{GG}_{1V} &= \frac{w\overline{gg}_{1V}}{\nabla_s} \\ &= \frac{\left[\rho_l g \frac{b^2 l}{8} \tan \phi \right] \left[\frac{b}{3} \tan \phi \right]}{\rho_s g \nabla_s} \\ &= \frac{\rho_l}{\rho_s} \frac{b^3 l}{12} \frac{\tan^2 \phi}{2 \nabla_s} \\ \overline{GG}_{1V} &= \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \frac{\tan^2 \phi}{2} \quad (6-30)\end{aligned}$$

or

$$\overline{GG}_{1V} = \overline{GG}_{1T} \frac{\tan \phi}{2}$$

which is to be expected, since the shift in center of gravity of the ship should be in the same direction as the shift in the center of gravity of the wedge of liquid (equation 6-14).

To determine the effect of the free surface on the righting arm, consider an expanded sketch of the $\overline{GZ}$ relationships in figure 6-18, in which the angle has been exaggerated for clarity of the relationships.

$$\begin{aligned}\overline{G_1Z_1} &= \overline{GZ} - \overline{GG}_{1T} \cos \phi - \overline{GG}_{1V} \sin \phi \\ &= \overline{GM} \sin \phi - \left(\frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \tan \phi \right) \cos \phi \\ &\quad - \left(\frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \frac{\tan^2 \phi}{2} \right) \sin \phi \\ \overline{G_1Z_1} &= \sin \phi \left[\overline{GM} - \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \left(1 + \frac{\tan^2 \phi}{2} \right) \right] \quad (6-31)\end{aligned}$$

However, the relationship

$$\overline{GZ} = \overline{GM} \sin \phi$$

is restricted to *small angles of inclination*, typically less than 7° to 10°, depending on the ship form. At 7° of heel,

$$\tan^2 \phi / 2 = 0.0075$$

and at 10° of heel,

$$\tan^2 \phi / 2 = 0.0155$$

Thus, for initial stability calculations the effect of the vertical shift in the center of gravity can be neglected, and equation 6-31 becomes

$$\overline{G_1Z_1} = \left(\overline{GM} - \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \right) \sin \phi \quad (6-32)$$

The term $\frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s}$ can be thought of as a negative correction to the metacentric height, and in particular as a virtual rise in the center of gravity of the ship, since M does not change.

In this case,

$$\overline{GG}_{\text{vert}} = \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s}$$

is the virtual rise of the center of gravity and is called the *free-surface correction*. This relationship will hold regardless of what liquid is contained in the tank, the location of the tank, and whether the ship is in fresh or salt water.

The free surface correction (*FSC*) should be applied to the calculation of the effective $\overline{GM}$ when free surfaces are known to exist in a ship. It should also be applied to the results of the inclining experiment discussed in the next section.

$$FSC = \frac{\gamma_l}{\gamma_s} \frac{i}{\nabla_s} \quad (6-33)$$

Thus,

$$\overline{GM}_{\text{eff}} = \overline{KM} - \overline{KG} - FSC \quad (6-34)$$

for small angles of inclination.

In the operational situations involving flooding damage, the tank or compartment may be considered flooded with water of the same density as that in which the ship floats. The relationship may then be simplified to

$$FSC = \frac{i}{\nabla_s} \quad (6-35)$$

Note that ∇_s is the *volume of displacement of the ship* and not the volume of the loose water.

Throughout this section, the analysis has assumed that the liquid in the tank did not come in contact with the top of the tank. Were this to happen, the magnitude of the weight shifts would be reduced by the effects of “pocketing,” as shown in figure 6-19. The influence of free surface can be reduced by filling all tanks to at least 95 percent full to cause pocketing to occur at lower angles of heel. The analysis of the free-surface effects at larger angles of heel is covered in chapters 8, 9, and 10.

Example 6-5: Calculate the effect of free surface on the oil barge in example 6-3, assuming the 80 tons of

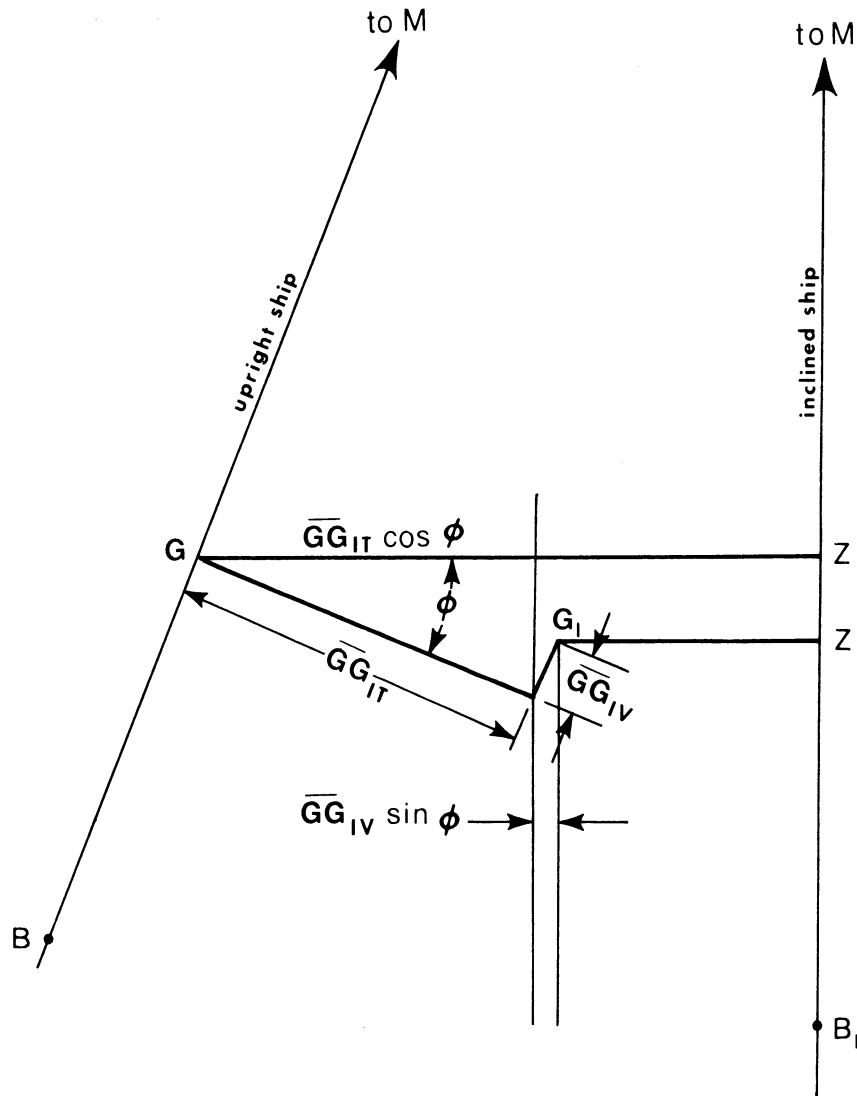


Figure 6-18. Effect of free surface on $\overline{GZ}$

oil (γ is equal to 0.86) do not completely fill tank 5. Also calculate the fully loaded $\overline{GM}$ and correct it for free surface if each compartment is not pressed full.

Solution: From example 6-3,

$$\overline{GM}_1 = 43.2 \text{ ft}$$

$$\nabla_1 = \Delta_1 \times 35 = 220 \times 35 = 7,700 \text{ ft}^3$$

$$FSC = \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla}$$

$$i = \frac{b^3 l}{12} = \frac{(17.5)^3 20}{12} = 8,932 \text{ ft}^4$$

$$FSC = \frac{0.86}{1.025} \frac{8,932}{7,700} = 0.97 \text{ ft}$$

$$\overline{GM}_{\text{eff}} = \overline{GM} - FSC$$

$$= 43.2 - 0.97$$

$$= \boxed{42.2 \text{ ft}}$$

With all the tanks filled but still having a free surface,

$$\Delta_2 = \Delta_0 + 800 = 940 \text{ tons}$$

$$\nabla_2 = 940 \times 35 = 32,900 \text{ ft}^3$$

$$T_2 = \frac{\nabla_2}{LB} = \frac{32,900}{100(35)} = 9.4 \text{ ft}$$

$$\overline{KG}_2 = \frac{\Delta_0 \overline{KG}_0 + \sum w \overline{Kg}}{\Delta_2}$$

$$= \frac{140(3.3) + 8(80)6.05}{940}$$

$$= 4.61 \text{ ft}$$

$$\overline{GM}_2 = \frac{T_2}{2} + \frac{B^2}{12T_2} - \overline{KG}_2$$

$$= \frac{9.4}{2} + \frac{35^2}{12(9.4)} - 4.61$$

$$= 10.95 \text{ ft}$$

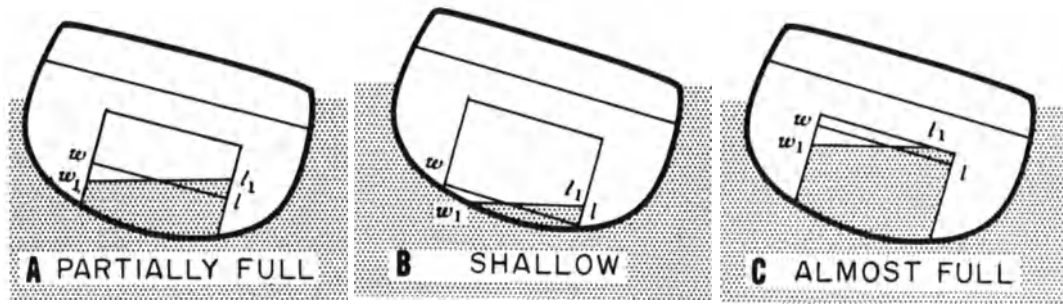


Figure 6-19. Pocketing of a free surface

$$\begin{aligned}
 \text{FSC} &= 8 \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla} \\
 &= 8 \frac{.86}{1.025} \frac{8,932}{32,900} \\
 &= 1.82 \text{ ft} \\
 \overline{GM}_{\text{eff}} &= 10.95 - 1.82 \\
 &= \boxed{9.1 \text{ feet}}
 \end{aligned}$$

6-9 Determination of the Center of Gravity: The Inclining Experiment

In preceding discussions, the importance of the location of the center of gravity was discussed in its relation to the metacenter and the center of buoyancy. Inasmuch as the locations of the center of buoyancy and the metacenter are given in relation to the keel, one must similarly locate the center of gravity in relation to the keel.

In the very early stages of design, the center of gravity is estimated from its actual position in similar ships. Later on in the design, the position is calculated by the long and tedious process of summing up all of the weights and moments of each item in the entire ship. This calculation is made not only to estimate the position of the center of gravity, but also to ensure that the total weight of the ship will not exceed the desired displacement.

The nature of the center of gravity should be kept in mind when one is thinking of its actual position. Remember that its location is not a function of the ship's underwater hull form, as are the locations of the metacenter and center of buoyancy. The position of G is a function of the distribution of the weights on board the ship and of the ship itself. Therefore, it is not possible to plot a curve showing the positions of the center of gravity for a range of drafts or displacements.

A ship's weight is customarily divided into the *light-ship displacement* and the variable load (deadweight). The light-ship displacement is defined as *the weight of the ship complete in every respect, including hull, machinery, outfit, equipment, water in the boilers at steaming level, and liquids in machinery and piping, but with all tanks and bunkers empty and no crew, passen-*

gers, cargo, stores, or ammunition on board. The later items comprise the variable load and equal the deadweight tonnage when the ship is floating at her greatest allowable draft (loaded displacement).

Before delivery to the ship owner, and following most ship alterations in which significant weights are added or removed, the center of gravity of the light ship is determined from an inclining experiment. The results of this experiment are calculated and tabulated in a booklet called the *Inclining Experiment Booklet* (U.S. Navy) or the *Stability Test Report* (U.S. Coast Guard). The centers of gravity of the variable loads are estimated from the arrangements and capacity plans for certain standard loading conditions. The center of gravity of the loaded ship can be computed from a tedious summation of moments, or a computer can be used to calculate the center of gravity for various loading conditions and the results tabulated in the *Stability and Loading Data Booklet* (U.S. Navy) or the *Trim and Stability Booklet* (U.S. Coast Guard). (Some large merchant ships with complicated cargo loads have on-board computers to indicate the current values of $\overline{KG}$ and $\overline{GM}$, as well as other stability characteristics, fore and aft drafts, and structural bending moments, as a function of the current loaded condition.)

THE INCLINING EXPERIMENT

The theory of the inclining experiment is based on the conditions requisite for equilibrium in an inclined position, as given by equation 6-18,

$$\Delta \overline{GM} \tan \phi = wt$$

Before the inclining experiment is performed, the values of w and t are determined. Knowing the athwartship distance t on deck available for the weight movement, an approximation of the size of the inclining weight w to produce an angle of inclination of a few degrees can be made by estimating $\overline{GM}$ from previous experiments with the same type of ship in a similar condition, or from the current estimate of $\overline{KG}$ for the ship to be inclined.

The inclining weights are generally ingots or pigs of iron, and are accurately weighed with the location of their centers of gravity plainly marked on them. They are

placed at known distances off the centerline. The total inclining weight w is generally made up in three or four lots and distributed at predetermined positions longitudinally along the deck. This makes possible several angles of inclination and also avoids concentration of too great a load on the deck structure.

Since the inclining experiment is often performed in the final construction phase, much outfitting equipment may be aboard. A thorough ship's survey should be made before inclining to estimate the percentage of unfinished work and the weight and location of items that will not be on board the finished ship. The items to be removed can be included in the calculation of the light ship $\overline{KG}$.

Several plumb bobs are suspended in positions such as hatch openings to afford a sufficient length of pendulum for accurate readings. The lengths of the plumb bobs must be such that the inclinations will produce deflections of at least 6 inches. The length is accurately measured just before the experiment.

The actual procedure is as follows:

1. The ship must be floating freely.
2. All dock lines should be slackened; all liquid tankage should be either full or empty insofar as possible, and the bilges dry so that there is a minimum of loose liquid. Boilers, wet machinery, and piping should have liquids at operating level, since this is the normal condition of operation at sea.
3. The experiment should be performed at slack tide, and when there is minimum wind. A preferable location is in a flooded graving dock with the caisson in position.
4. All weights aboard should be secure so they will not shift. Checks should be made on the contents of all storerooms.
5. The crew should be sent ashore, or if aboard, should be restricted to some convenient location that is made a matter of record.

6. After the inclining weights have been put aboard, and immediately prior to commencing the inclining, the following data are recorded:

- a) drafts forward and aft and any residual list
- b) temperature of water
- c) density of water by hydrometer

From these data and the displacement and other curves, the displacement Δ may be determined accurately.

7. When all is ready, one or more of the inclining weights is moved according to the previously arranged program, and upon signal after movement is completed, observers read and record:

- a) deflection of pendulums
- b) amount of inclining weight and distance moved

The deflection of the pendulum divided by the fixed pendulum length equals the tangent of the angle of inclination ϕ . From the $\tan \phi$, along with values of w and t , the metacentric height $\overline{GM}$ is calculated. Usually several inclinations are made to both port and starboard

with intermediate checks at zero. In such cases, the tangents of the angles of inclination from the initial position are plotted against the moment (both port and starboard) of the inclining weights from their initial position. Theoretically, the plot of tangents of small angles of inclination against the corresponding inclining moments will be a straight line. Variation of the resulting plot from a straight line indicates that conditions for the experiment are not favorable or that an error has been made. The slope of the straight line fitted to these points is datum to solve for the metacentric height $\overline{GM}$. The straight-line plot is a means of averaging the data graphically. The slope can also be determined from a linear regression analysis on a hand calculator, but the plot is crucial in checking for errors. From the plot,

$$\text{Slope} = \frac{\tan \phi}{\text{inclining moment}}$$

Thus,

$$\overline{GM} \text{ (as inclined)} = \frac{wt}{\Delta \tan \phi} = \frac{1}{\Delta \text{slope}} \quad (6-36)$$

Now, from the curves of form at the inclining experiment draft, take the value for the height of the metacenter above the keel $\overline{KM}$ and subtract the value of $\overline{GM}$ obtained from the inclining experiment. The final result is the height of the center of gravity above the keel $\overline{KG}$ for the ship, as inclined.

The $\overline{GM}$ calculated from equation 6-36 includes the effect of the inclining weights and associated gear on the $\overline{KG}$ of the ship as well as the influence of any free surfaces that are not part of the normal operating condition of the ship. The light ship $\overline{KG}_0$ can be determined by using the weight-removal equation 6-10, along with the free surface correction for tanks known to be neither full nor empty.

$$\overline{KG}_1 = \overline{KM}_1 - \overline{GM} \text{ (as inclined)} - \sum \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla_s} \quad (6-37)$$

Then

$$\overline{KG}_0 = \frac{\overline{KG}_1 \Delta_1 - \sum w \overline{Kg} \text{ (as inclined)}}{\Delta_0} \quad (6-38)$$

where subscript "1" refers to the as-inclined condition and subscript "0" refers to the light-ship condition. Equation 6-38 can be used to calculate the light-ship condition. The $\overline{KG}$ for the operating condition can then be computed by adding in the normal load of fuel,

water, supplies, personnel, etc., at the appropriate vertical locations.

$$\overline{KG} \text{ (operating)} = \frac{KG_0\Delta_0 + \sum w\overline{Kg}}{\Delta \text{ (operating)}} + FSC \quad (6-39)$$

The following is an example of an inclining experiment.

Example 6-6: Given: A DD 692-class destroyer, whose draft readings give a mean draft of 13 feet 9 inches with no trim, is inclined as follows:

Moment	Tangent of angle of inclination
160 ft tons stbd	.0220
320 ft tons stbd	.0460
0 ft tons —	.0020
160 ft tons port	.0180
320 ft tons port	.0420

The free surface correction has been estimated at 0.21 feet and the specific gravity of the water by hydrometer is 1.020. The inclining gear weighs 38 tons with a $\overline{Kg}$

of 25 feet. Determine the $\overline{KG}$ as inclined and the $\overline{KG}$ without the inclining gear.

Solution: The curves of form for the DD 692 at a draft of 13 feet 9 inches give a value of 19.3 feet for $\overline{KM}$ and a displacement in fresh water of 3,175 tons.

$$\text{corrected } \Delta = 1.020(3,175) = 3,238 \text{ tons}$$

The inclining data is plotted in figure 6-20 and a graphical solution yields

$$\text{slope} = \frac{0.108}{800 \text{ ft-tons}} = 1.35 \times 10^{-4}/\text{ft-ton}$$

In this case, because there is little data scatter, a least-squares analysis (linear-regression analysis) on a calculator gives the same slope. This latter technique may be used to supplement the graphical method when there is data scatter to yield a more accurate answer.

Using equation 6-36,

$$\begin{aligned} \overline{GM} \text{ (as inclined)} &= \frac{1}{\Delta \text{slope}} = \frac{1}{3,238 \times 1.35 \times 10^{-4}} \\ &= 2.29 \text{ ft} \end{aligned}$$

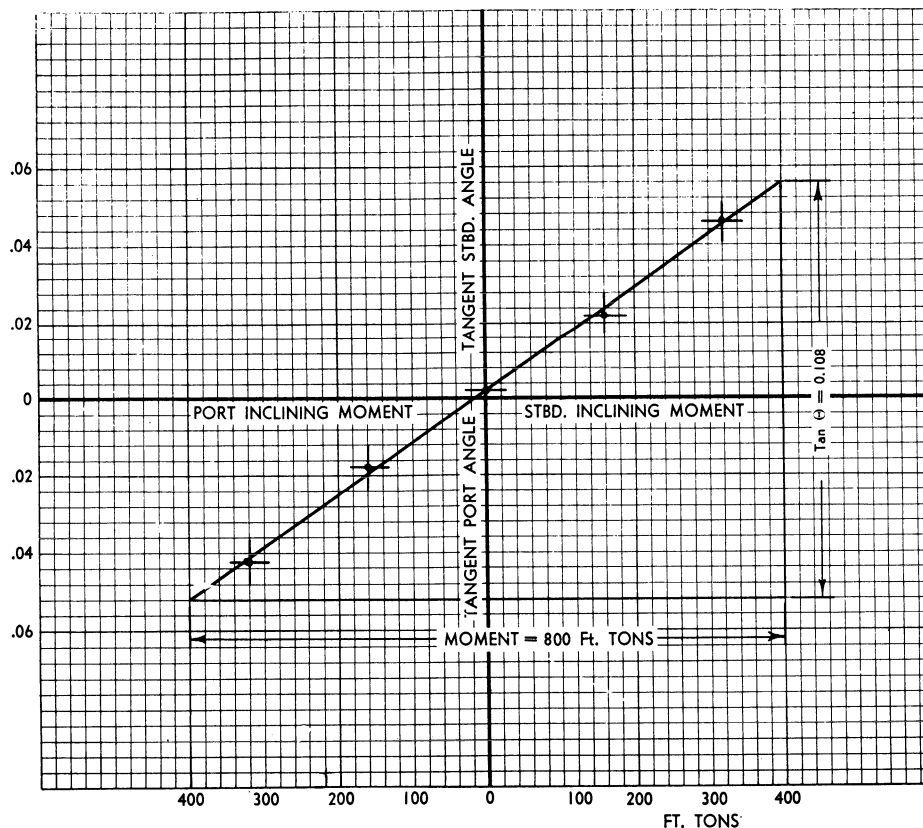


Figure 6-20. Plot of tangents of angles vs. inclining moments for example problem

Thus, in the inclined condition,

$$\begin{aligned}\overline{KG}_1 &= \overline{KM}_1 - \overline{GM}_1 - FSC \\ &= 19.3 - 2.29 - 0.21 \\ &= 16.8 \text{ ft}\end{aligned}$$

And the $\overline{KG}_0$ without the inclining gear is

$$\begin{aligned}KG_0 &= \frac{\Delta_1 \overline{KG}_1 - wKg}{\Delta_0} \\ &= \frac{3,238(16.8) - 38(25)}{3,200} \\ &= \boxed{16.7 \text{ feet}}\end{aligned}$$

This is not the final light-ship condition, but the analysis illustrates the method for calculating the various centers of gravity.

Problems

6-1. Rework example 6-1 using Simpson's rule of integration and compare answers.

6-2. Plot the variation of $\overline{GM}$ with values of γ from 0 to 1.0 for example 6-2.

6-3. Consider a homogeneous rectangular block of wood as in example 6-2. What ratio of width (beam) to molded depth will insure positive initial stability regardless of the specific gravity of the wood?

6-4. Consider a box-shaped lighter in which the $\overline{GM}$ is to be held at 25 percent of the draft. Plot the ratio of $\overline{KG}/T$ as a function of beam-to-draft ratio.

6-5. Consider a box-shaped lighter for which the ratio of $\overline{KG}/T$ is to be maintained at unity. Plot the ratio of $\overline{GM}/T$ as a function of beam-to-draft ratio.

6-6. Compare the initial stability of a triangular hull form (in body plan) with a waterline beam of 12 feet, a draft of 6 feet, and a $\overline{GM}$ of 2 feet with that of a rectangular barge having the same length, beam, $\overline{GM}$, and displacement. Determine the maximum value of $\overline{KG}$ for each hull form.

6-7. A rectangular barge 120 feet long and 36 feet wide is floating upright in fresh water with a draft of 9 feet. There is a 40-ton weight on deck 14 feet above the keel. How far must this weight be moved transversely to produce a 3° list to starboard? The $\overline{KG}$ of the loaded barge is equal to 10 feet.

6-8. A DD 692-class destroyer at a draft of 14 feet 6 inches fore and aft in salt water has a $\overline{KG}$ of 15.5 feet. How far must a 40-ton weight 20 feet above the keel be moved athwartships to produce a 3° list to port?

6-9. A landing craft used to transport tanks for the Normandy invasion during World War II was 126 feet

long, 45 feet wide, and 12 feet deep. Nine tanks, each weighing 70 tons, were arranged in three rows with the center of gravity of the port and starboard rows being 11 feet off the centerline. The rectangular barge, when empty, displaced 990 tons and had a $\overline{KG}$ of 4 feet. The center of gravity of each of the tanks was located 15 feet above the keel. What angle of list would the landing craft assume if all the port-side tanks were off-loaded first?

6-10. A ship has a displacement of 7,200 tons, a $\overline{KM}$ of 26.4 feet, a $\overline{KG}$ of 22 feet, and a 3° starboard list. How much water (in tons) must be pumped from a tank whose center of gravity is 16 feet to starboard of the centerline to a tank whose center of gravity is 20 feet to port of the centerline to remove the list? Ignore any rise in the center of gravity of the water.

6-11. A ship has a displacement of 35,000 tons, a $\overline{KM}$ of 36 feet, a $\overline{KG}$ of 32 feet, and a 2° list to starboard. Two hundred tons of fuel are consumed from a tank whose center of gravity is 15 feet to port of the centerline and whose $\overline{KG}$ is 3.0 feet. $\overline{KM}_1$ is equal to 36.1 feet after the weight removal. Determine the residual angle of list.

6-12. A large bulk carrier, 700 feet long with a beam of 100 feet and a draft of 36 feet, is to be dry-docked. Assume that the ship is wall sided and has a C_B of 0.9 and a C_{IT} of 0.84. Determine the value of $\overline{KG}$ below which the ship will be stable throughout the dry-docking operation as long as it is partially waterborne.

6-13. A DD 692-class destroyer is preparing to enter dry dock for a major overhaul. Just before entering the dry dock, $\overline{KG}$ is determined to be 15.5 feet. The drafts fore and aft are equal to 14 feet 6 inches, and there is no list. What will be the level of water above the keel of the ship when the transverse metacentric height becomes zero?

6-14. Consider the oil barge used in examples 6-3 and 6-5. Calculate the free-surface correction and the fully loaded $\overline{GM}_{\text{eff}}$ if each compartment is not pressed full but

a) the barge is divided by two longitudinal bulkheads into 15 equal-sized tanks, 20 feet by 11.67 feet.

b) the barge is divided by three longitudinal bulkheads into 20 equal-sized tanks, 20 feet by 8.75 feet.

6-15. A fleet oiler's wing tank is 90 feet long, 20 feet wide, and 30 feet deep. It contains 44,000 cubic feet of oil (specific gravity of 0.87). The oiler's displacement is 30,000 tons in salt water (specific gravity of 1.025). What percent of the oiler's effective $\overline{GM}$ of 4.0 feet can be attributed to the free surface in the wing tank?

6-16. Calculate the angle of list of the box-shaped

lighter in example 6-3, assuming tank 5 has a free surface.

6-17. An inclining experiment is performed on a rectangular barge 140 feet long and 36 feet wide. It displaces 1,728 tons in salt water. When a weight of 50.35 tons is shifted 12 feet to starboard, the barge lists to starboard by 4°. Determine $\overline{KG}$ for the barge as inclined.

6-18. A ship displacing 7,000 tons undergoes an inclining experiment. A free-surface correction has been estimated at 0.25 feet and the $\overline{KM}$ at that draft equals 24.3 feet. The following data was recorded:

<i>Moment</i>	<i>Tan of angle of inclination</i>
0 foot tons	0.0018
400 foot tons, port	-0.0185
800 foot tons, port	-0.0410
0 foot tons	0.0018
400 foot tons, starboard	0.0215
800 foot tons, starboard	0.0435

The inclining gear weighs 60 tons with a $\overline{Kg}$ of 30 feet. Determine the $\overline{KG}$ as inclined and the $\overline{KG}$ without the inclining gear.

Longitudinal Stability and Trim

7-1 Trim

Most ships are many times longer than wide and, as a result, are much more resistant to changes in longitudinal inclination than in transverse inclination. Transverse inclinations are generally described both statically and dynamically in terms of list, heel, or roll angles. Longitudinal inclinations of a ship at rest, however, are described in terms of changes in trim, defined in chapter 3 as the difference in drafts, fore and aft. This approach is taken for several reasons, including the facts that trim angles are generally quite small and that most ships have draft marks, fore and aft, that can be used (with corrections to the actual location of the draft marks) as a measure of trim.

$$\text{trim} = T_A - T_F \quad (7-1)$$

where T_A = draft at the after perpendicular
 T_F = draft at the forward perpendicular exclusive of appendages such as sonar domes.

Note that the definition of trim implies that positive trim is associated with trim by the stern and negative trim is associated with trim by the bow. In practice, however, “by the stern” and “by the bow” may be used instead of a sign convention.

Most naval vessels are designed to operate “on an even keel,” meaning that the designed drafts, fore and aft, are equal to the mean draft at midships.

$$T_M = \frac{T_A + T_F}{2} \quad (7-2)$$

Many commercial vessels, including high speed craft and tugboats with large propellers, are frequently designed with a greater draft aft than forward to ensure adequate propeller submergence. This is called *drag*.

$$\text{drag} = (T_A - T_F)_{\text{design}} \quad (7-3)$$

For such ships, the trim is defined as *the difference between the after and forward drafts in excess of the drag*.

$$\text{trim} = (T_A - T_F) - \text{drag} \quad (7-4)$$

The trim angle θ for small changes in trim is found from equation 6-20.

$$\theta = \tan^{-1} \frac{(T_A - T_F)}{L} \quad (7-5)$$

where the length L is equal to the distance between the fore and aft perpendiculars L_{PP} or the distance between draft marks, as appropriate. These trim relationships are illustrated in figure 7-1 for a ship designed on an even keel (drag is equal to zero). Note that the ship trims about the longitudinal center of flotation (LCF), which is not located at midships except for vessels with longitudinal symmetry, such as rectangular barges. Most naval vessels are fuller in the stern area than in the bow, so the centroid of the waterplane area on the designer’s waterline is aft of amidships. The reason that the ship trims about the center of flotation is that for a constant displacement, the volume of the immersed wedge of water WFW_1 aft must equal the volume of the emerged wedge LFL_1 forward. This can only happen if the center of rotation F is located at the centroid of the waterplane area, which is called the *longitudinal center of flotation* (LCF).

7-2 Longitudinal Initial Stability

The location of the longitudinal metacenter M_L is established using the same approach as that used for transverse stability. The height of the longitudinal metacenter above the keel $\overline{KM}_L$ is found from

$$\overline{KM}_L = \overline{KB} + \overline{BM}_L \quad (7-6)$$

where $\overline{KB}$ is the same as for the transverse case. The longitudinal metacentric radius $\overline{BM}_L$ is found by sum-

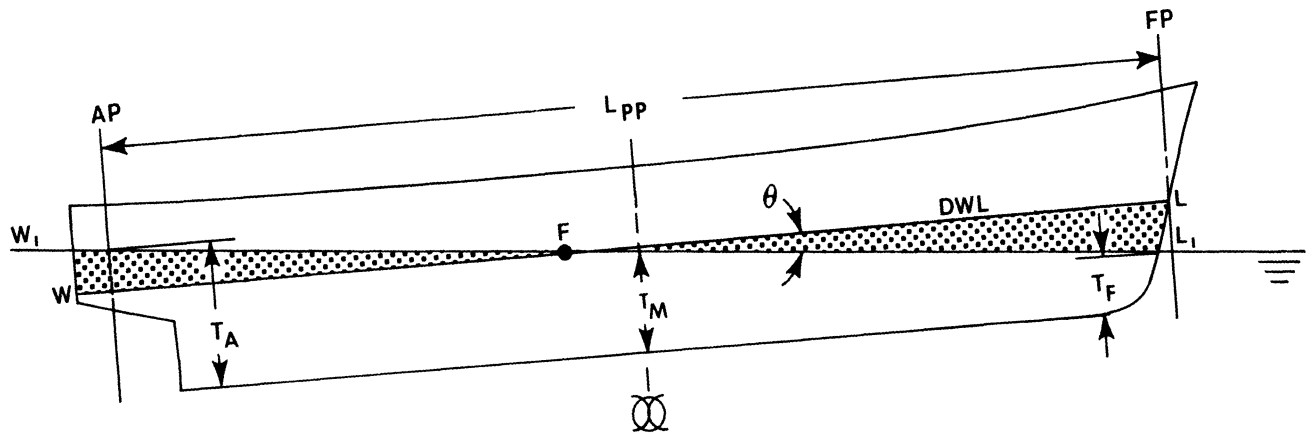


Figure 7-1. Longitudinal trim

ming moments about the center of flotation of the immersed and emerged wedges (figure 7-2) and equating the sum to the righting moment caused by the shift in the center of buoyancy of the ship. By analogy with the development in the previous chapter, one obtains

$$\overline{BM}_L = \frac{I_L}{\nabla} \quad (7-7)$$

where I_L = the moment of inertia of the longitudinal waterplane about the centroid of the waterplane area.

The longitudinal metacentric height is then found from equation 3-14.

$$\overline{GM}_L = \overline{KM}_L - \overline{KG}$$

Note that on the curves of form for actual ships, values of $\overline{BM}_L$ are plotted directly and must be added to the value of $\overline{KB}$ at a particular draft to obtain $\overline{KM}_L$.

7-3 Moment to Change Trim

Any longitudinal weight movement in a ship results in a change in the longitudinal center of gravity of the ship, similar to the effect of transverse weight shifts but much smaller in magnitude. As illustrated in figure 7-2, equating couples produced by the weight movement with the couple produced by the shift in the center of gravity of the ship results in

$$wl = \Delta \overline{GG}_1 \quad (7-8)$$

But $\overline{GG}_L$ equals $\overline{GM}_L \tan \theta$, so that

$$wl = \Delta \overline{GM}_L \tan \theta \quad (7-9)$$

Since trim was defined as T_A minus T_F , the change in trim may be defined as

$$\delta \text{trim} = \delta T_A - \delta T_F \quad (7-10)$$

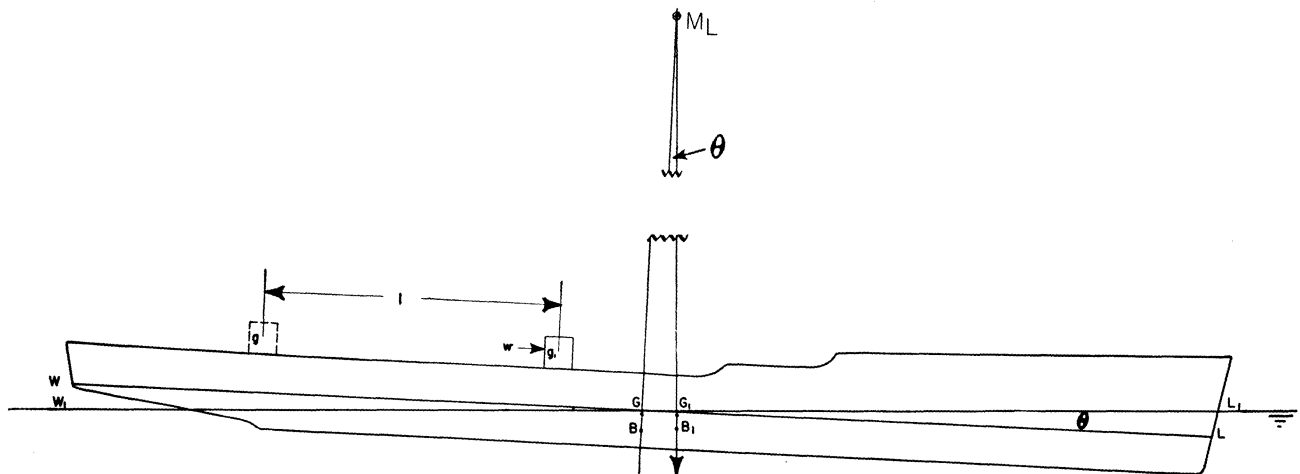


Figure 7-2. Longitudinal trimming moments

Combining equation 7-9 with equations 7-5 and 7-10, one obtains

$$\delta \tan \theta = \frac{\delta \text{trim}}{L} = \frac{wl}{\Delta \overline{GM}_L} \quad (7-11)$$

To simplify calculations of small changes in trim for a given vessel, it is common to combine the influence of displacement $\overline{GM}_L$ and length for a small change in trim, say one inch, and define the moment to change trim one inch $MT1''$ as

$$MT1'' = wl = \Delta \overline{GM}_L \frac{1/12}{L}$$

or

$$MT1'' = \frac{\Delta \overline{GM}_L}{12L} \quad (7-12)$$

$\overline{GM}_L$ and $\overline{BM}_L$ are both quite large numbers for actual ships, and the difference between $\overline{KB}$ and $\overline{KG}$ is a small percentage of $\overline{BM}_L$. Since $\overline{BM}_L$ is a function of ship geometry, it is common to define an approximate $MT1''$ for ships in salt water and include it in the curves of form (see curve 12 in the back pocket of the book).

$$\begin{aligned} MT1'' (\text{approx}) &= \frac{\Delta \overline{BM}_L}{12L} \\ &= \frac{1}{12L} \frac{\nabla}{35} \frac{I_L}{\nabla} = \frac{I_L}{420L} \end{aligned} \quad (7-13)$$

Thus the approximate change in trim may be calculated using the curves of form.

$$\delta \text{trim} = \frac{\text{trimming moment}}{MT1''} \quad (7-14)$$

Example 7-1: Compare $MT1''$ with the approximate value using $\overline{BM}_L$ for a DD 692 destroyer with drafts, fore and aft, in salt water of 13 feet 6 inches and a $\overline{KG}$ of 15 feet.

Solution: From curves of form

$$\overline{KB} = 810 \times \frac{1}{100} = 8.1 \text{ ft (curve 3)}$$

$$\overline{BM}_L = 455 \times \frac{200}{100} = 910 \text{ ft (curve 11)}$$

$$\Delta = 3,180 \text{ tons (curve 1)}$$

$$L = 383 \text{ ft}$$

$$\overline{KM}_L = \overline{KB} + \overline{BM}_L = 8.1 + 910 = 918 \text{ ft}$$

$$\overline{GM}_L = \overline{KM}_L - \overline{KG} = 918 - 15 = 903 \text{ ft}$$

$$\begin{aligned} MT1'' &= \frac{\Delta \overline{GM}_L}{12L} = \frac{3,180(903)}{12(838)} \\ &= \boxed{625 \text{ ft-tons/in}} \end{aligned}$$

$$\begin{aligned} MT1'' (\text{approx}) &= \frac{\Delta \overline{BM}_L}{12L} = \frac{3,180(910)}{12(383)} \\ &= \boxed{630 \text{ ft-tons/in}} \end{aligned}$$

Curve 12 at T_M is equal to 13 feet 6 inches and gives

$$MT1'' = 635 \times \frac{100}{100} = \boxed{635 \text{ ft-tons/in}}$$

All of the answers are within the useful range of accuracy obtainable from the curves of form.

Example 7-2: A rectangular barge 144 feet long by 36 feet wide and 10 feet high displaces 864 tons in fresh water at level trim. The center of gravity of the barge is 10 feet above the keel. Determine

a) The longitudinal and transverse metacentric radii and their ratio

$$\overline{BM}_L = \frac{I_L}{\nabla} = \frac{L^3 B}{12 L B T} = \frac{L^2}{12 T}$$

$$T = \frac{\nabla}{LB} = \frac{36 \Delta}{LB} = \frac{36(864)}{144(36)} = 6 \text{ ft}$$

$$\overline{BM}_L = \frac{144^2}{12(6)} = \boxed{288 \text{ feet}}$$

$$\overline{BM} = \frac{I_T}{\nabla} = \frac{B^3 L}{12 L B T} = \frac{B^2}{12 T}$$

$$\overline{BM} = \frac{36^2}{12(6)} = \boxed{18 \text{ feet}}$$

$$\frac{\overline{BM}_L}{\overline{BM}} = \frac{288}{18} = 16 = \left(\frac{L}{B}\right)^2$$

b) The moment to change trim one inch $MT1''$

$$MT1'' = \frac{\Delta \overline{GM}_L}{12L}$$

$$\overline{GM}_L = \overline{KB} + \overline{BM}_L - \overline{KG}$$

$$= \frac{6}{2} + 288 - 9 = 282 \text{ ft}$$

$$MT1'' = \frac{864(282)}{12(144)} = \boxed{141 \text{ ft-tons/in}}$$

c) The distance a weight of 30 tons would have to be moved to produce a trim of one foot by the stern

$$\delta \text{trim} = \frac{wl}{MT1''}$$

$$\begin{aligned} l &= \frac{\delta \text{trim} \times MT1''}{w} \\ &= \frac{12 \text{ in} \times 141 \text{ ft-tons/in}}{30 \text{ tons}} \\ &= \boxed{56.4 \text{ ft}} \end{aligned}$$

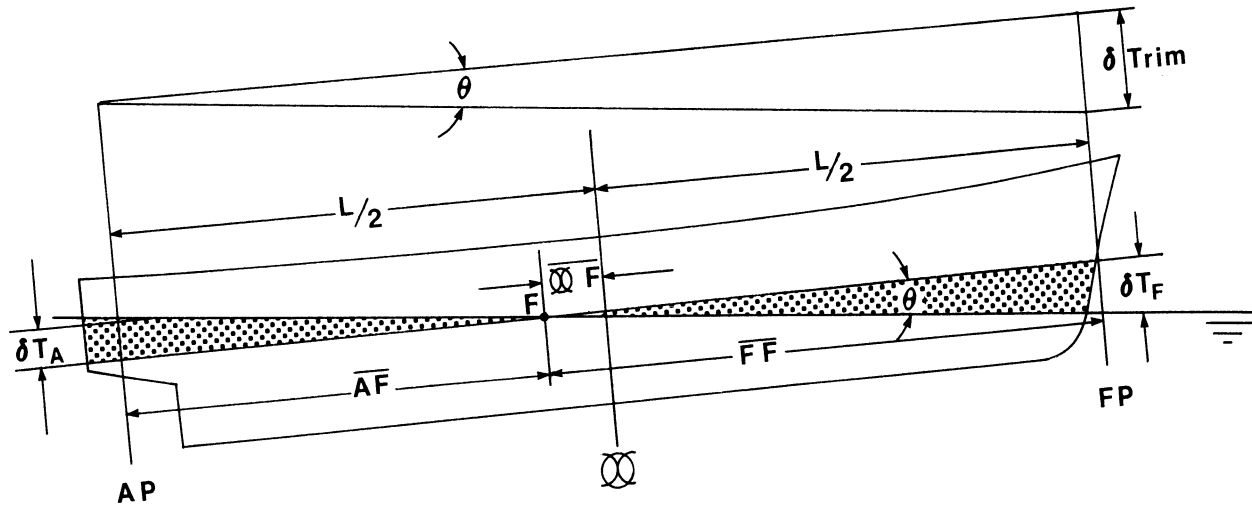


Figure 7-3. Change in trim

7-4 Change in Drafts, Fore and Aft, as a Result of Trim Changes

Since ships trim about a longitudinal center of flotation that is generally not located amidships, the total trim change, δtrim , must be apportioned using similar triangles between the change in draft aft δT_A and the change in draft forward δT_F , as illustrated in figure 7-3.

$$\delta \tan \theta = \frac{\delta \text{trim}}{L} = \frac{\delta T_A}{AF} = \frac{-\delta T_F}{FF} \quad (7-15)$$

where AF = distance from the after perpendicular to the center of flotation F

$$= L/2 + OF$$

FF = distance from the forward perpendicular to the center of flotation F

$$= L/2 - OF$$

OF = distance from the midship station O to the LCF (curve 6 on the DD 692 curves of form). The sign of OF is positive when the LCF is located ahead of the midships station O and negative if F is aft of O .

In general, a longitudinal weight shift will change the drafts, fore and aft, as follows:

$$\delta T_A = \delta \text{trim} \frac{AF}{L} \quad (7-16a)$$

$$= \frac{wl}{MT1''} \left(\frac{L/2 + OF}{L} \right) \quad (7-16b)$$

$$\delta T_F = \delta T_A - \delta \text{trim} \quad (7-17)$$

On ships with an initial trim, the change in drafts caused by a longitudinal weight shift may be determined

by first calculating the mean draft from the given drafts, fore and aft, and finding the corresponding $MT1''$ from the curves of form. If the displacement Δ is needed in the problem, correct the displacement corresponding to the mean draft for the addition to displacement due to one foot change of trim by the stern (curve 13 on the DD 692 curves of form). Note that the mean draft is not quite equal to the draft at the center of flotation. Thus, if the problem statement gives a mean draft and a stated trim by the bow or by the stern, the actual drafts, fore and aft, are found by adding or subtracting one half of the trim to the mean draft as appropriate. The initial trim centered around the draft at amidships should not be combined with the change in trim based on the center of flotation to determine the new drafts, fore and aft. Rather, one should calculate the change in trim, fore and aft, from equations 7-16 and 7-17 and add them algebraically to the initial drafts, fore and aft. This method automatically corrects the actual mean draft for the effects of trim on displacement, provided that the magnitude of the trim is within the linear range of the curves of form. The method is illustrated in the following example.

Example 7-3: A DD 692-class destroyer is at sea with drafts, fore and aft, of 13 feet one inch and 13 feet 11 inches. It is desired to bring the ship to an even keel by transferring fuel oil from a tank located 90 feet aft of midships to a tank located 70 feet forward of midships. Determine the drafts, before and after the transfer, the displacement of the ship, and the amount of oil that must be transferred to bring the ship to an even keel.

The original mean draft T_{M_o} equals

$$T_{M_o} = \frac{T_A + T_F}{2} = \frac{13'11.0'' + 13'1.0''}{2} = 13'6.0''$$

The displacement of the ship is found from curves 1 and 13 on the curves of form.

$$\Delta = 3,180 LT + 995 \frac{LT}{ft} \times \frac{2}{100} \times \frac{10 \text{ in}}{12 \text{ in/ft}}$$

$$\Delta = 3,180 + 16.6 = \boxed{3,197 \text{ tons}}$$

The amount of oil is found from equation 7-14 and curve 12.

$$MT1'' = 635 \text{ ft-tons/in}$$

$$\text{trimming arm } l = 90 + 70 = 160 \text{ ft}$$

$$\begin{aligned} \text{trimming moment } wl &= \delta \text{trim} \times MT1'' \\ &= -10'' \times 635 \text{ ft-tons/in} \\ &= -6,350 \text{ ft-tons (by the bow)} \end{aligned}$$

$$\begin{aligned} \text{oil transfer } w &= \frac{wl}{l} \\ &= \frac{6,350 \text{ ft-tons}}{160 \text{ ft}} \\ &= \boxed{39.7 \text{ tons}} \end{aligned}$$

The change in trim, fore and aft, requires the use of curve 6 to determine the location of the LCF and equation 7-16.

$$\overline{\text{LF}} = 1,125 \text{ tons} \times \frac{2 \text{ ft}}{100 LT} = -22.5 \text{ ft (aft of } \overline{\text{LF}})$$

$$\begin{aligned} \delta T_A &= \delta \text{trim} \frac{\overline{AF}}{L} \\ &= -10 \text{ in} \frac{(383/2 - 22.5)}{383} \\ &= -4.4 \text{ in} \end{aligned}$$

$$\begin{aligned} \delta T_F &= \delta T_A - \delta \text{trim} \\ &= -4.4 - (-10) \\ &= 5.6 \text{ in} \end{aligned}$$

Thus the new drafts, fore and aft, are

$$\begin{aligned} T_{F1} &= T_{F0} + \delta T_F \\ &= 13'1.0'' + 5.6'' \end{aligned}$$

$$T_{F1} = 13'6.6''$$

$$\begin{aligned} T_{A1} &= T_{A0} + \delta T_A \\ &= 13'11.0'' - 4.4'' \end{aligned}$$

$$T_{A1} = 13'6.6''$$

and

$$\begin{aligned} T_{M1} &= \frac{13'6.6'' + 13'6.6''}{2} \\ &= 13'6.6'' \end{aligned}$$

Note that the mean draft has increased by 0.6 inches. This is necessary to maintain the displacement of 3,197 tons at level trim. This can be verified by three methods.

First, using the tons-per-inch immersion from curve 7,

$$\text{TPI} = 1,410 \text{ tons} \times \frac{2}{100} = 28.2 \text{ tons/in}$$

$$\begin{aligned} \delta \Delta &= 0.6 \text{ in} \times 29.2 \text{ tons/in} \\ &= 16.9 \text{ tons} \end{aligned}$$

which is essentially the same correction as was obtained from curve 13 for the addition to displacement caused by the initial 10-inch trim by the stern.

The second method involves estimating the "effective mean draft" for level trim at a displacement of 3,197 tons. The draft read from curve one is difficult to estimate more accurately than to the nearest half inch, since the DD 692 curves of form have a draft increment of 3 inches. One might use a value of 13 feet 6.5 inches

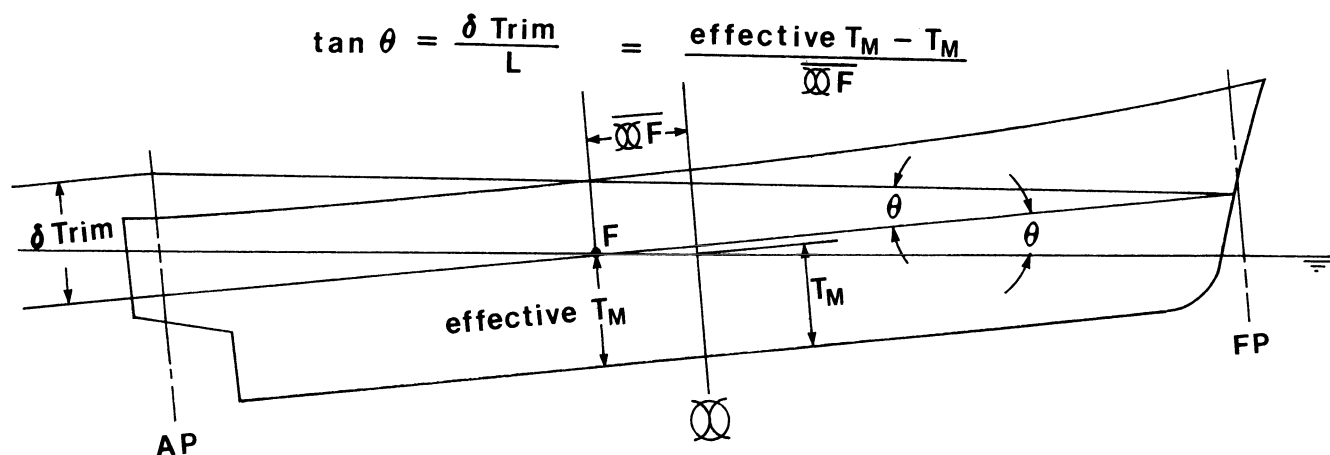


Figure 7-4a. Effective mean draft (down by the stern)

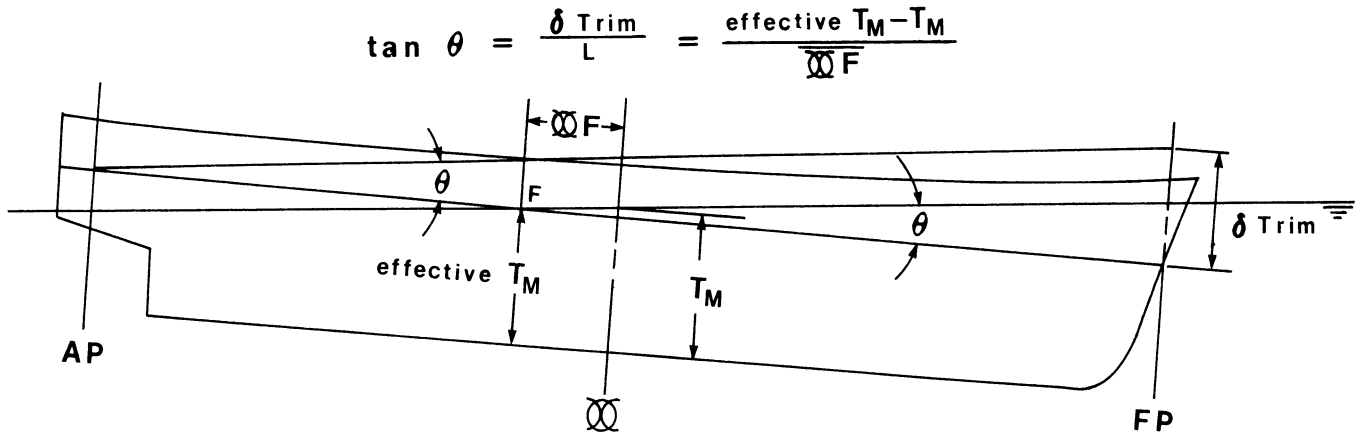


Figure 7-4b. Effective mean draft (down by the bow)

from curve one or, alternately, combine the effects of curve 13 and curve 7 as follows:

$$\begin{aligned} \text{effective } T_M &= T_{M_o} - \frac{\delta \Delta}{\text{TPI (curve 7)}} \quad (7-18) \\ &= 13'6.0'' - \left(\frac{-16.6}{28.2} \right) \\ &= 13'6.6'' \text{ as before} \end{aligned}$$

The sign on $\delta \Delta$ is taken from the direction change to level trim.

The effective mean draft can also be obtained from the draft beneath the center of flotation F . By similar triangles, illustrated in figures 7-4a and 7-4b,

$$\begin{aligned} \text{effective } T_M &= T_{M_o} + \overline{\mathbb{O}F} \frac{\delta \text{trim}}{L} \quad (7-19) \\ &= 13'6.0'' + \frac{(-22.5 \text{ ft})(-10 \text{ in})}{383 \text{ ft}} \\ &= 13'6.0'' + 0.6'' \\ &= 13'6.6'' \end{aligned}$$

Thus from equations 7-18 and 7-19 the curve 13 correction to displacement is based on

$$\delta \Delta = \frac{-\overline{\mathbb{O}F}(\delta \text{trim})\text{TPI}}{L} \quad (7-20)$$

If the concept of effective mean draft is used, then an alternate method for calculating changes in draft for ships with an initial trim is possible. First, calculate the trimming moment required to give the ship its initial trim from equation 7-14, and add it algebraically to the trimming moment calculated from the longitudinal weight shift. The combined trimming moment is then used with equations 7-16 and 7-17 to determine the change in drafts, fore and aft. In the example problem, these

trimming moments would cancel, since the initial trimming moment would be 6,350 ft-tons and the trimming moment was generated to trim down by the bow. If the new condition involved trim by the bow, the combined trimming moment would be negative and curve 13 would have to be used twice, along with equation 7-18, for a double correction to the effective mean draft. Equation 7-19 requires only the calculation of the total trim change.

7-5 Effect of Weight Addition/Removal on Draft and Trim

In the previous section on transverse and longitudinal weight shifts, the heeling or trimming moment was based on the distance moved by the center of gravity of the weights. As in the case for the effect of weight addition on transverse stability, a specific reference point is needed to calculate the effect of weight addition on draft and trim. Since an arbitrary weight addition causes both an increase in displacement and a trimming moment, it is useful to replace the effect of an off-axis weight addition with a force, acting at the center of rotation, and a couple, equal to the product of the force and the distance from the center of rotation to the actual location of the weight. Since the center of rotation about which a ship trims is the longitudinal center of flotation (LCF), a weight added at this location will cause a "parallel sinkage" at both bow and stern, provided the mean center of the added buoyant layer b is reasonably close to the initial and final centers of flotation.*

* The center of the added buoyant layer is sometimes referred to as the *mean center of flotation* or the *center of immersion*. The curves of form indicate the longitudinal position of the centers of flotation as centers of gravity of waterplanes aft of $\mathbb{O}$. An arithmetical mean of the locations of the centers of flotation of the original and final waterlines will be considered as the center of the added buoyant layer, even though the variation of the center of flotation with draft is not one of the constant rate. In many cases, the actual difference in longitudinal position between the centers of flotation of two relatively close waterlines is so small that it may be neglected. However, this is a matter that must be determined in each individual problem depending upon the limits of accuracy required, the range of drafts involved, etc.

The sequence involved in calculating the effects of weight addition are illustrated in figures 7-5, 7-6, and 7-7. The weight to be added forward in figure 7-5 is first added at the mean center of the added buoyant layer b , which is taken as the average of the initial and final positions of the LCF (figure 7-6).

$$\overline{b} = \frac{\overline{F_1} + \overline{F_0}}{2} \quad (7-21)$$

This causes a parallel sinkage δT_M (not to be confused with the effective-mean-draft correction in the previous section).

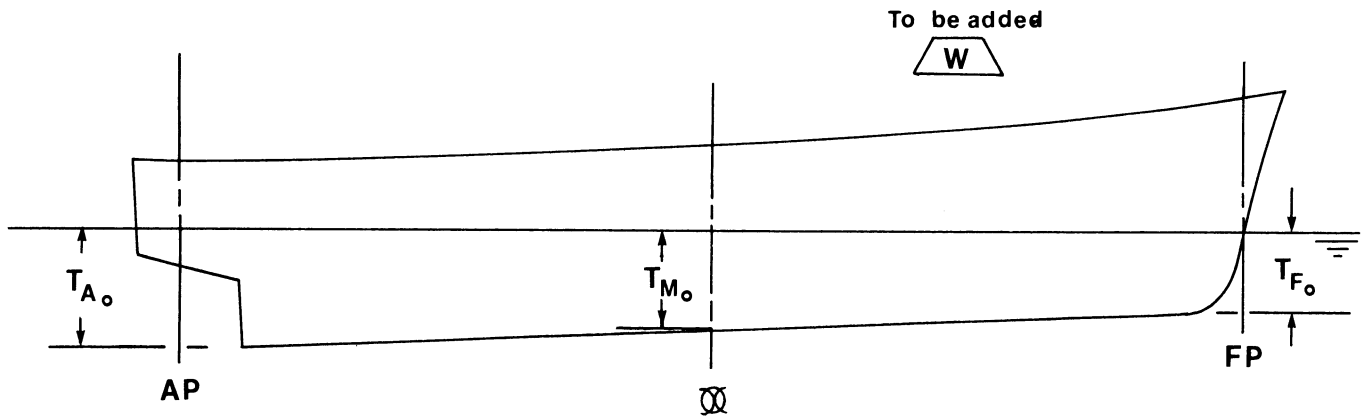


Figure 7-5. Original condition

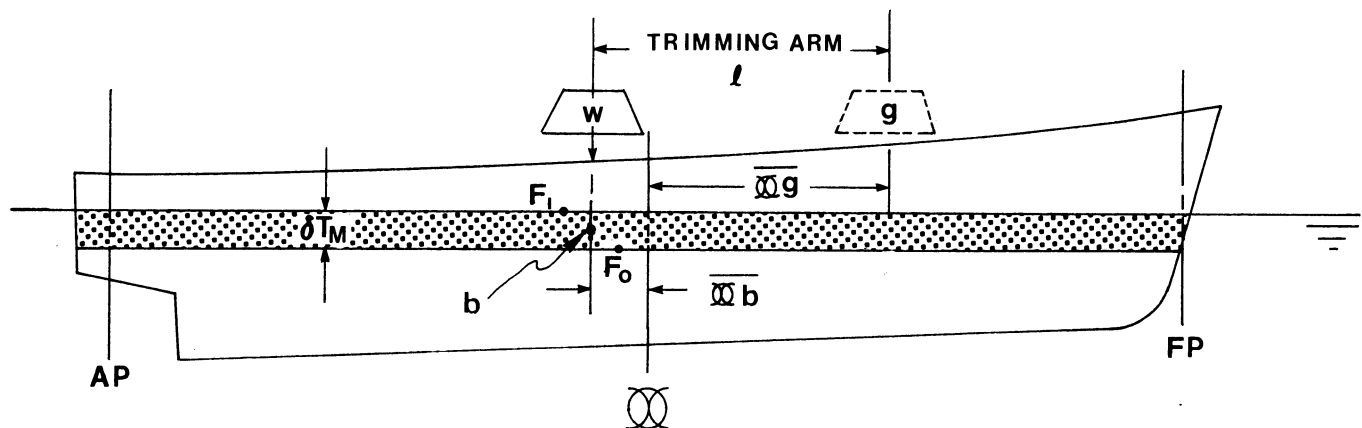


Figure 7-6. Weight added (parallel sinkage)

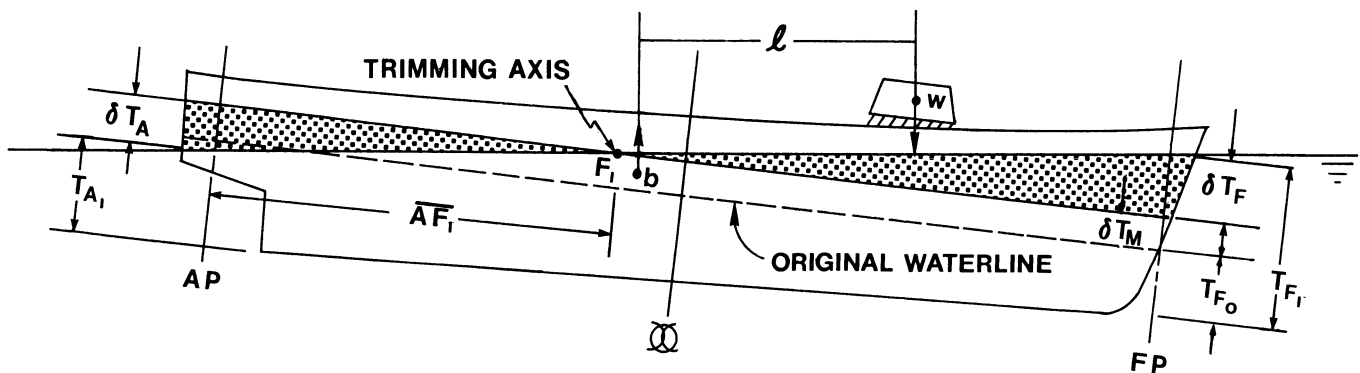


Figure 7-7. Weight added (trim)

$$\delta T_M = \frac{w}{\text{TPI}} \quad (7-22)$$

Equation 7-14 is used to calculate the change in trim from the $MT1''$ and the trimming moment, based on the distance from the mean LCF to the longitudinal location of the weight. Since both locations will frequently be specified relative to midships,

$$\text{trimming arm } l = \overline{\text{OB}} - \overline{\text{OG}} \quad (7-23)$$

where $\overline{\text{OG}}$ = the distance from midships to the location of the added weight.

The signs of $\overline{\text{OG}}$ and $\overline{\text{OB}}$ are positive for locations forward of O and negative if they are aft of O .

$$\delta \text{trim} = \frac{w(\overline{\text{OB}} - \overline{\text{OG}})}{MT1''} \quad (7-24)$$

and the sign of δtrim is positive for trim by the stern. For multiple weight additions

$$\delta \text{trim} = \frac{\overline{\text{OB}} \sum w - \sum (w \overline{\text{OG}})}{MT1''} \quad (7-24a)$$

The change in drafts, fore and aft, caused by the trimming moment are calculated using equations 7-16 and 7-17. The final drafts, fore and aft, are then calculated from

$$T_{F_1} = T_{F_0} + \delta T_M + \delta T_F \quad (7-25)$$

$$T_{A_1} = T_{A_0} + \delta T_M + \delta T_A$$

where the sign on δT_m is positive for weight additions and negative for weight removals.

In order to avoid storing separate calculations on a calculator, the equations can be combined.

$$T_{A_1} = T_{A_0} + \frac{w}{\text{TPI}} + \delta \text{trim} \left(\frac{L/2 + \overline{\text{OF}}_1}{L} \right) \quad (7-26)$$

$$T_{F_1} = T_{F_0} + \frac{w}{\text{TPI}} - \delta \text{trim} \left(\frac{L/2 - \overline{\text{OF}}_1}{L} \right)$$

associating a negative w with weight removal.

Example 7-4: A DD 692-class destroyer operating off Alaska experiences severe icing topsides. One hundred seventy tons of ice are distributed more heavily on the windward side (port) than on the leeward side. Assume the center of gravity of the ice is 25 feet above the keel, 8 feet to port of the centerline of the ship, and 45 feet forward of amidships. Before the icing occurred, the ship had a mean draft of 13 feet 6 inches with level trim and a $\overline{KG}$ of 15 feet.

a) Determine the new $\overline{KG}$, the new $\overline{GM}$, and the angle of list. From the curves of form when T_M is 13'6",

$$\Delta_0 = 3,180 \text{ tons}$$

$$\Delta_1 = 3,180 + 170 = 3,350 \text{ tons}$$

$$\text{TPI} = 1,410 \times \frac{2}{100} = 28.2 \text{ tons/in}$$

Parallel sinkage is determined as follows:

$$\delta T_M = \frac{w}{\text{TPI}} = \frac{170}{28.2} = 6 \text{ in}$$

At

$$T_M = 14'$$

$$\overline{KM}_1 = 960 \times \frac{2}{100} = 19.2 \text{ ft}$$

$$\begin{aligned} \overline{KG}_1 &= \frac{\Delta_0 \overline{KG}_0 + w \overline{Kg}}{\Delta_1} \\ &= \frac{3,180(15) + 170(25)}{3,350} \\ &= \boxed{15.5 \text{ feet}} \end{aligned}$$

$$\begin{aligned} \overline{GM}_1 &= \overline{KM}_1 - \overline{KG}_1 \\ &= 19.2 - 15.5 \\ &= \boxed{3.7 \text{ feet}} \end{aligned}$$

$$\tan \phi = \frac{wl}{\Delta_1 \overline{GM}_1} = \frac{170(8)}{3,350(3.7)} = 0.11$$

$$\text{list } \phi = \boxed{6.3^\circ \text{ to port}}$$

b) Determine the new mean draft and the drafts, fore and aft, for the icing condition. (Assume the angle of list does not affect the draft.) To find the new mean draft,

$$MT1'' = 640 \text{ ft-tons/in at } T_M = 14'$$

$$\overline{\text{OF}}_0 = 1,125 \times \frac{2}{100} = -22.5 \text{ ft}$$

$$\overline{\text{OF}}_1 = 1,120 \times \frac{2}{100} = -22.4 \text{ ft}$$

$$\overline{\text{OB}} = \frac{\overline{\text{OF}}_0 + \overline{\text{OF}}_1}{2} = -22.45 \text{ ft}$$

$$\text{trimming arm } l = \overline{\text{OB}} - \overline{\text{OG}}$$

$$= -22.45 - 45$$

$$= -67.45 \text{ ft}$$

$$\delta \text{trim} = \frac{wl}{MT1''} = \frac{170(-67.45)}{640}$$

$$= -17.9 \text{ in (by the bow)}$$

(Note that $\overline{\text{OB}}$ had too many significant figures, but they were carried until δtrim was rounded off.)

From equations 7-16 and 7-17,

$$\begin{aligned}\delta T_A &= \delta \text{trim} \frac{\overline{AF}}{L} = \delta \text{trim} \left(\frac{L/2 + \overline{OF}_1}{L} \right) \\ &= -17.9 \left(\frac{383/2 - 22.4}{383} \right) \\ &= -7.9 \text{ in} \\ \delta T_F &= \delta T_A - \delta \text{trim} \\ &= -7.9 - (-17.9) \\ &= 10 \text{ in}\end{aligned}$$

The final drafts, fore and aft, are found using equation 7-25.

$$\begin{aligned}T_{F_1} &= T_{F_0} + \delta T_M + \delta T_F \\ &= 13'6'' + 6'' + 10'' \\ &= \boxed{14 \text{ feet } 10 \text{ inches}} \\ T_{A_1} &= T_{A_0} + \delta T_M + \delta T_A \\ &= 13'6'' + 6'' - 7.9'' \\ &= \boxed{13 \text{ feet } 4.1 \text{ inches}} \\ T_{M_1} &= \frac{T_{F_1} + T_{A_1}}{2} = \frac{14'10'' + 13'4.1''}{2} \\ &= \boxed{14 \text{ feet } 1 \text{ inch}}\end{aligned}$$

which is an additional one inch beyond the parallel sinkage, since the ship is now trimmed by the bow.

Problems

7-1. A rectangular barge 150 feet long, 35 feet wide, and 14 feet high is loaded to a mean draft of 8 feet in salt water. Assume the initial $\overline{KG}$ is 5 feet.

- What is the initial displacement of the barge?
- A 300-ton load is added whose center of gravity is 6 feet above the deck (20 feet above the keel). The load is distributed so that the barge is in level trim with no list. Determine the new draft.
- Determine the new $\overline{KG}_1$.
- Determine the new $\overline{GM}_1$ and $\overline{GM}_{L_1}$ (transverse and longitudinal metacentric heights).
- If bad weather causes the 300-ton load to shift 5 feet to starboard and 12 feet forward of amidships, what will be the angle of list?
- Based on the conditions stated in (e) above, what are the new drafts fore and aft?
- Estimate the remaining freeboard at the starboard bow.

7-2. A DD 692-class destroyer originally in level trim in salt water draws 13 feet forward and 15 feet aft as a result of the addition of 150 tons of emergency supplies.

- What is the total displacement of the ship?
- Where is the longitudinal center of gravity of the emergency supplies relative to amidships?

7-3. A destroyer of the DD 692 class has a mean draft of 13 feet 3 inches and is trimmed 8 inches by the stern. The original $\overline{KG}$ is 15.5 feet. Three hundred tons of fuel oil is taken on in a tank whose center of gravity is 100 feet aft of amidships and 8 feet above the keel on the longitudinal centerline. Determine the new $\overline{GM}$ and the new drafts, fore and aft.

7-4. A 550-ft-long ship displaces 5,000 tons and has a TPI of 50 tons/in, MT1" of 750 ft-tons/in, and a center of flotation 35 feet aft of amidships. One hundred tons of fuel are transferred from a fuel tank 75 feet forward of amidships to a tank 75 feet aft of amidships. If the draft forward was 22 feet 6 inches and the draft aft 21 feet 6 inches before the transfer, determine the drafts fore and aft following the transfer. What is the change in mean draft?

7-5. A ship has a displacement of 10,000 tons, an LCF 60 feet aft of amidships, MT1" of 1,600 ft-tons/in, TPI of 60 tons/in, length of 600 feet, draft aft of 25 feet, and draft forward of 23 feet. Six hundred tons of fuel are added 12 feet aft of amidships. Determine the drafts fore and aft following the fuel addition and note the change in mean draft.

7-6. A DD 993 displacing 7,000 tons at a mean draft of 18 feet takes on 150 tons of ammunition and missiles at an average distance of 106 feet forward of amidships. The ship has a length of 529 feet, a beam of 55 feet, an LCF 42 feet aft of amidships, a TPI of 50 tons/in, and an MT1" of 1,480 ft-tons/in. If the ship was initially in level trim, determine the new drafts, fore and aft, and the change in mean draft.

7-7. A large aircraft carrier displacing 79,500 tons at a mean draft of 35 feet 4 inches has a length of 990 feet, a TPI of 230 tons/in, and a MT1" equal to 12,900 ft-tons/in. Ten F-14s each weighing 60,480 pounds are moved an average distance of 564 feet from the bow to the stern and eight A-6s each weighing 44,800 pounds are moved aft an average distance of 500 feet. The initial drafts were 35 feet 8 inches forward and 35 feet 0 inches aft. The LCF is 56 feet aft of amidships. Determine the new drafts, fore and aft, and the change in mean draft.

7-8. A DD 692-class destroyer is floating with no list in salt water at drafts of 12 feet 6 inches forward and 13 feet 3 inches aft, with a $\overline{KG}$ of 15.8 feet. A

weight of 150 tons is placed 25 feet above the keel, 41 feet forward of midship, and 5 feet to starboard of centerline. Determine or calculate the following:

- a) original displacement
- b) new displacement
- c) parallel sinkage

d) change of trim

e) new drafts forward and aft

f) $\overline{GM}$

g) angle of heel (approximately)

Show the source of all numbers used and all calculations to obtain your answer.

General Stability at Large Angles of Heel

8-1 Initial Stability and Range of Stability

In describing stability characteristics, the terms *stiff* and *tender* are commonly used to compare ships of similar type and size. A so-called stiff ship is one that has a short, jerky roll; it is difficult to heel initially, or in other words, has large initial righting arms. Such a ship would then also have a large $\overline{GM}$, since $\overline{GM}$ is a measure of the initial stability and an indication of a high rate of development of initial righting moments. *There is frequently the misunderstanding that stiff ships are essentially more stable and safe than tender, or easily heeled, ships.*

The fallacy of this conception can be seen by comparing a curve of righting arm versus angle of heel (introduced in section 6-3) for a typical destroyer with that for an offshore supply vessel constructed before the stability criteria were changed in 1964. Figure 8-1 illustrates that although the $\overline{GM}$ of the offshore supply vessel is twice that of the destroyer, the range of stability is much less. A destroyer, built for high speed and maneuverability, has a relatively narrow beam and a modest metacentric height. The freeboard of a destroyer is quite high in comparison with its other dimensions, which contributes to the ship's relatively large maximum righting arm and range of stability. Because of the gentle initial slope of a destroyer's stability curve, it must be expected that the ship will roll in a seaway to such angles where the larger righting arms develop.

The offshore supply vessel, on the other hand, has a large beam-to-draft ratio combined with a relatively shallow draft, which gives it a large $\overline{GM}$ and a stiff response in a seaway compared to other vessels its size. However, the low freeboard that surrounds the cargo area causes the deck edge to "go under" at moderate angles of heel, reducing its range of stability to an unusually low value for an offshore vessel. This reduced

range of stability resulted in the loss of at least eight supply vessels in severe seas in the Gulf of Mexico before the Coast Guard modified the stability criteria for such vessels to include righting-arm considerations in line with the discussion in section 8-9.

The $\overline{GM}$ of passenger vessels is generally very low in consideration of the comfort of the passengers. But the range of stability on the other hand is usually large, which can be attributed to the high freeboard.

The primary yardstick for the safety of a ship is neither the $\overline{GM}$ nor the range of stability, but the maximum righting arm and the angle at which this maximum arm occurs. But also, to say that these values are unrelated to $\overline{GM}$ and range would not be wholly true. Usually the stability characteristics of a ship are such that a large $\overline{GM}$ indicates a large maximum righting arm and a large angle of maximum righting arm indicates a large range of stability.

Combining the above considerations, they can be summed up simply as follows: *The most desirable stability characteristics for ships are those that combine an adequate maximum righting arm at an adequate angle of inclination with a substantial range of stability.*

8-2 Overall Stability

The most satisfactory means of presenting a complete picture of stability is a plot of the righting arms with their angles of inclination for several displacements, each of which is called a static-stability curve. Such a curve may be used to determine several important characteristics for each displacement, among which are (1) the righting arm at any inclination, (2) the $\overline{GM}$, (3) the angle of maximum righting moment, (4) the range of stability, and (5) the dynamic stability (related to the area under the righting-arm curve). The righting mo-

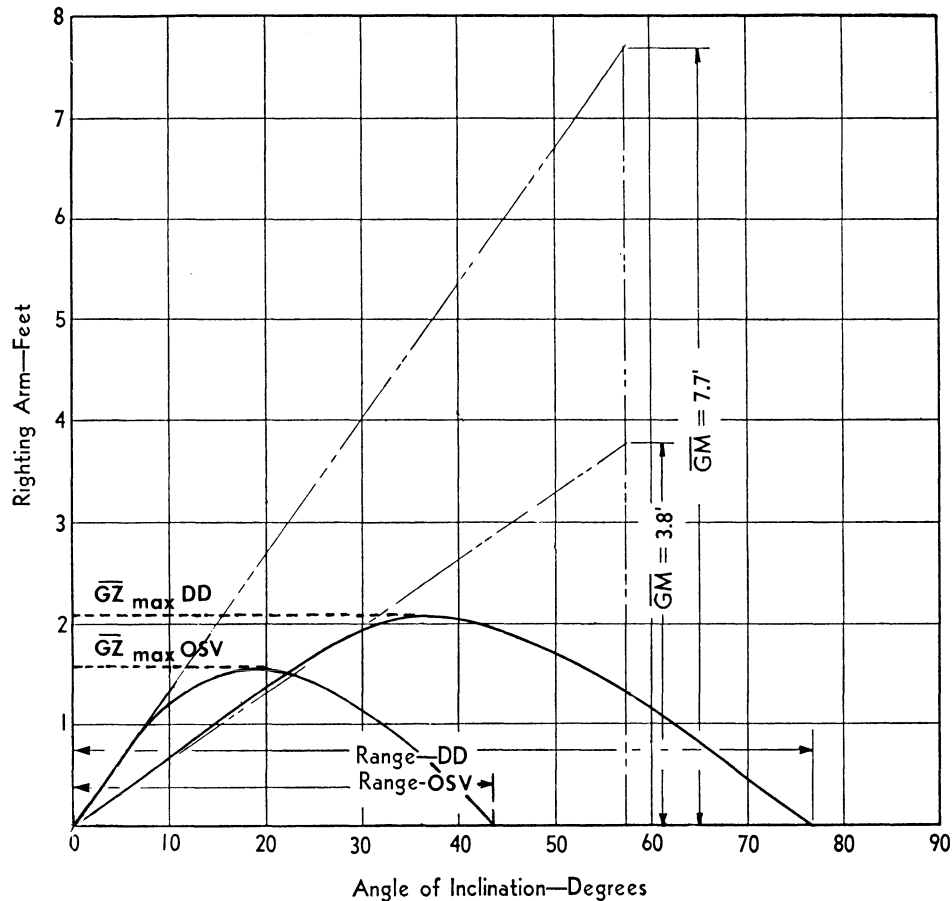


Figure 8-1. Comparison of stability curves for a destroyer (DD) and an offshore supply vessel (OSV)

ment at each angle of inclination equals the righting arm times the displacement of the ship.

It is possible to compute the righting arms ($\overline{GZ}$) for a range of angles of inclination, but this involves a tedious process of locating the many positions of B over the range of inclination. There are several methods for determining $\overline{GZ}$ by calculation or mechanical integration. A brief description of one of the mechanical-integration methods follows.

DEVELOPMENT OF CROSS CURVES OF STABILITY

A double-body plan showing the shape of both sides of each section of the ship is drawn. Figure 8-2 is a double-body plan in which the even-numbered sections have been omitted in the interest of clarity. An integrator, which is an instrument for measuring areas and moments of areas simultaneously, is now adapted to the following procedure.

The double-body plan is placed under an integrator in such a way that the axis of the instrument is perpendicular to the inclined waterline, for which the righting arm is to be determined (W_1L_1 , corresponding to up-

right waterline WL_3 in this case), and passes through a selected point on the vertical centerline (A). The tracing point of the instrument is then run successively over the underwater portion of each section, and the area and moment of area of each section about the integrator axis is recorded. The underwater portion of section 9 below the inclined waterline W_1L_1 is indicated by the shaded outline. The areas and moment of areas are integrated over the waterline length by the trapezoidal rule or Simpson's rule. The integration of areas is the underwater volume beneath the inclined waterline. This volume, when divided by the specific volume (35 ft³/ton), gives the displacement beneath the inclined waterline. The integration of moment of area is the moment of volume about a plane through the instrument axis and perpendicular to the sections. The moment of volume divided by the volume gives the distance of the center of buoyancy from the integrator axis. Note that the force of buoyancy acts in a line parallel to the instrument axis, inasmuch as the latter was set up perpendicular to the inclined waterline. The buoyant force is shown by the dashed arrow in figure 8-2.

If one assumes that the ship's center of gravity is

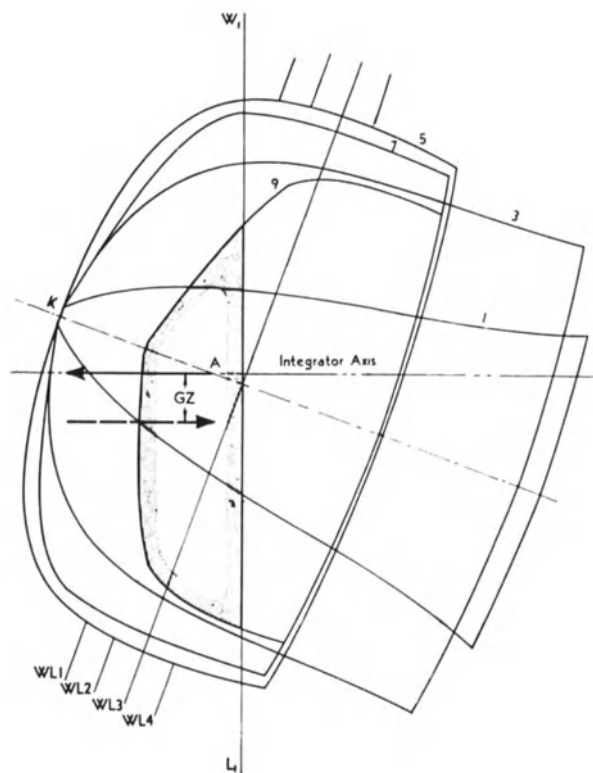


Figure 8-2. Partial double-body plan showing use of integrator for calculating GZ

located at the intersection of the vertical centerline and the instrument axis (point A in figure 8-2), then the line of action of the gravitational force will coincide with the integrator axis, as shown by the full arrow. The distance between the line of action of the center of buoyancy and the integrator axis is the righting arm GZ . This is, of course, for the given angle of inclination at the calculated displacement and for the assumed position of the center of gravity.

This calculation is made for angles of inclination from 0° through 90° in 10° increments. The process is repeated for the same inclinations about other upright waterlines, such as WL_1 , WL_2 , and WL_4 of figure 8-2. Note that this requires thirty-six calculations, nine for varying angles about each of four upright waterlines. Note further that in considering any upright waterline, such as WL_3 , the underwater volume (and displacement) will vary with the angle of inclination. It should be understood that the displacement of a ship will not change simply because the ship is inclined. However, for the purpose of computation, it is far simpler to maintain the same set of waterlines, taking into account that there will be a different displacement below waterline WL_3 when the drawing is inclined to 10° than when inclined to 20° . The result of each calculation is a righting arm for a specified angle of inclination and at a calculated displacement.

The results of the multitudinous calculations obtained from the procedure as described above are the righting arms of the ship for every specified angle of inclination (at 10° increments through 90°) at various displacements.

These values are plotted as curves, with righting arms as ordinates and displacements as abscissa, with an individual curve for each angle of inclination considered. These curves are called the *cross curves of stability*, a typical example of which is shown in figure 8-3. Figure 8-3 notes that the axis is assumed to be 14 feet above the bottom of the keel amidships. This indicates that the axis of the integrator used to develop the cross curves intersected the centerline 14 feet above the bottom of the keel. Hence, the assumed $\overline{KG}$ for these curves is 14 feet for this particular ship (a DD 692-class destroyer).

Although the technique just described is still used by practicing naval architects, it has been largely replaced by the use of numerical methods on computers, as described in appendix B. The U.S. Navy currently uses a program called SHCP (Ship Hull Characteristics Program) to calculate the cross curves of stability as well as the hydrostatic curves of form, the floodable-length curves, the damaged-stability calculations, and the longitudinal-strength calculations, all from a single set of digitized body-plan sections and other required inputs.

CURVES OF STATIC STABILITY

Consider for a moment the cross curves in three dimensions instead of two, with the angle of inclination as the third coordinate. The combination of cross curves and stability curves produces a three-dimensional surface, such as the one in figure 8-4. Now, selecting a particular displacement, take a section through this figure. The resulting curve is a plot of the righting arm as ordinate and the angle of inclination as abscissa. In other words, this sectional curve is a curve of static stability. This curve shows the righting arm at any angle of inclination for the original assumed position of the center of gravity and the displacement selected.

The value of the cross curves is emphasized in view of the variation in displacement in everyday operation. The cross curves, therefore, are the source for the static-stability curve at any operating displacement.

If it is desired to plot a static-stability curve for any specified displacement, the values of the righting arms are taken from the cross curves for each angle of inclination directly above the desired displacement of the abscissa of the cross curves. These values are plotted on the static-stability coordinates. Figure 8-5 gives the static-stability curve for a ship at 3,300 tons displacement at the assumed $\overline{KG}$ of 14 feet.

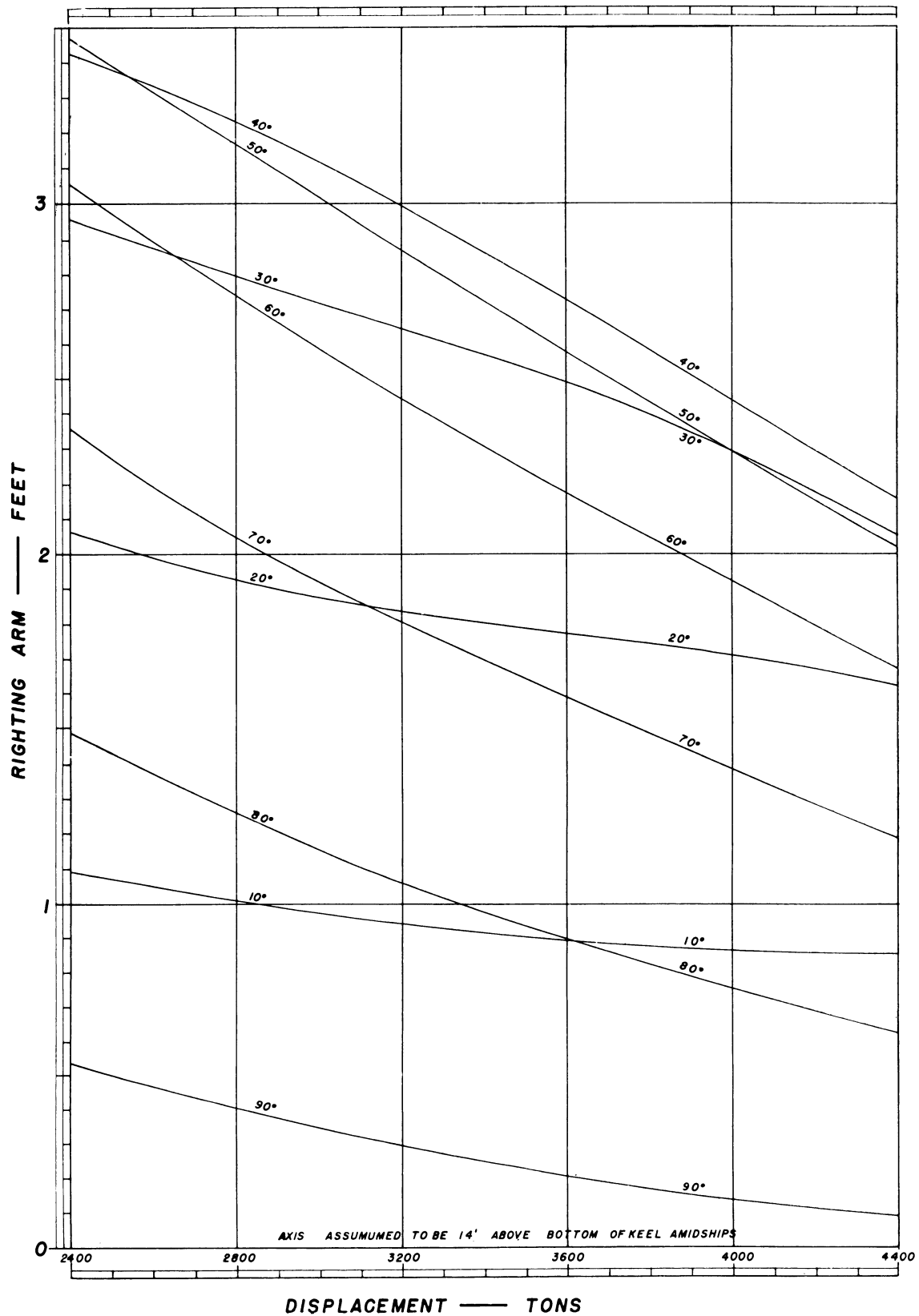


Figure 8-3. Cross curves of stability for a typical destroyer-type ship (DD 692)

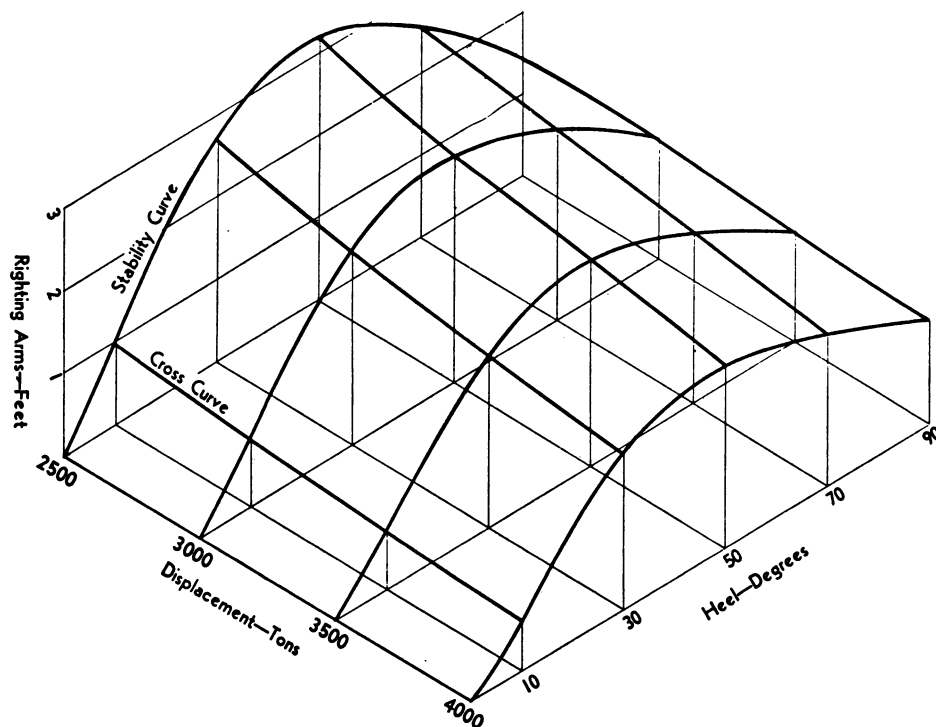


Figure 8-4. Cross curves and stability curves

8-3 Vertical Correction for Position of G

It will be remembered that the position of the center of gravity was assumed in the original computation of righting arms for the cross curves. After the exact location of the center of gravity and the displacement corresponding to each of the standard conditions of loading have been determined by the inclining experiment, final static-stability curves are drawn for the standard conditions. The plotting of these curves in-

volves a correction for the distance between the actual and assumed positions of the center of gravity.

For example, suppose the assumed position of the center of gravity had been placed on the designer's waterline at A in figure 8-6. Now, for this displacement and the actual condition of loading, suppose the true position of the center of gravity is at G , above the assumed position by the distance $\overline{AG}_V$, where the subscript denotes a vertical change.

If one were to incline the ship to some finite angle

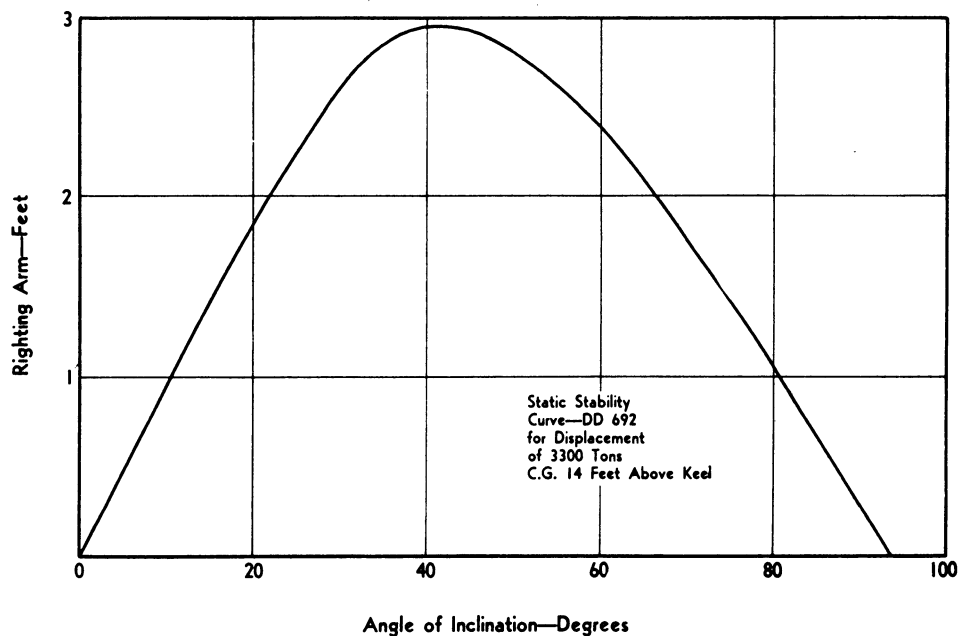


Figure 8-5. Uncorrected static-stability curve

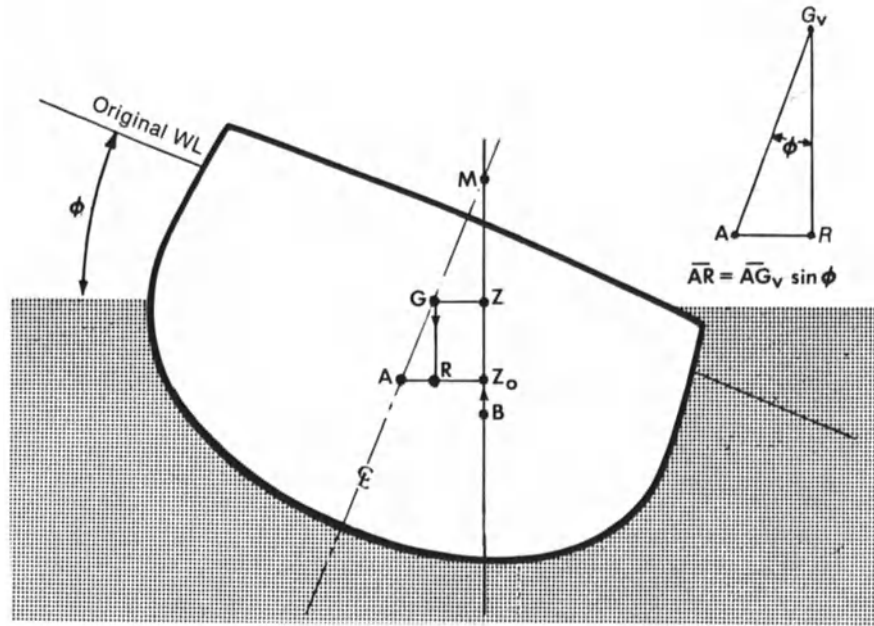


Figure 8-6. Loss of righting arm due to a rise in the center of gravity

ϕ , the actual righting arm will be $\overline{GZ}$ instead of $\overline{AZ}_0$, as it would have been if the center of gravity had been at A, as was originally assumed.

From the figure,

$$\overline{GZ} = \overline{AZ}_0 - \overline{AR}$$

but

$$\overline{AR} = \overline{AG}_v \sin \phi$$

$$\therefore \overline{GZ} = \overline{AZ}_0 - \overline{AG}_v \sin \phi \quad (8-1)$$

This means that when the real center of gravity is above the assumed position, any righting arm for the final corrected curve is equal to the righting arm for the uncorrected curve, minus the distance between the real and assumed centers of gravity, multiplied by the sine of the angle of inclination.

If the actual position of G is below the assumed position A, it will be found by similar reasoning that the correction ($\overline{AG}_v \sin \phi$) is additive. It is necessary to actually apply such a correction to the original stability curve to obtain the final corrected stability curve for any selected displacement.

$\overline{AG}_v \sin \phi$ may be plotted as a sine curve, with the maximum ordinate $\overline{AG}_v$ at 90° , to the same scale as the original stability curve. A convenient practice is to plot $\overline{AG}_v \sin \phi$ above the abscissa axis when G is above A and below the abscissa axis when G is below A. The sine curve may be thought of then as a new axis for the stability curve, and the final stability curve is always that portion of the original curve above the sine correction curve. Note that the corrected righting arm for any angle of inclination is measured by the vertical distance between the uncorrected curve and the sine curve (figures 8-7 and 8-8).

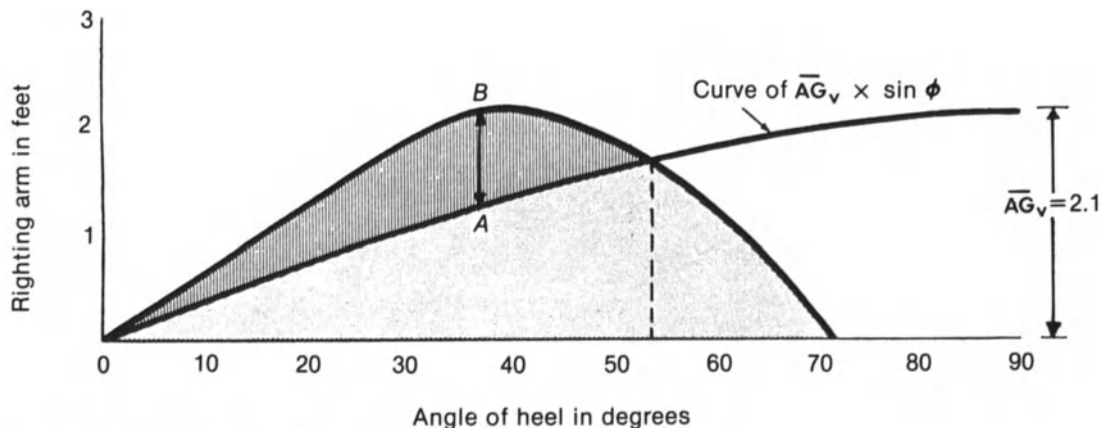


Figure 8-7. Sine curve superimposed on the original stability curve

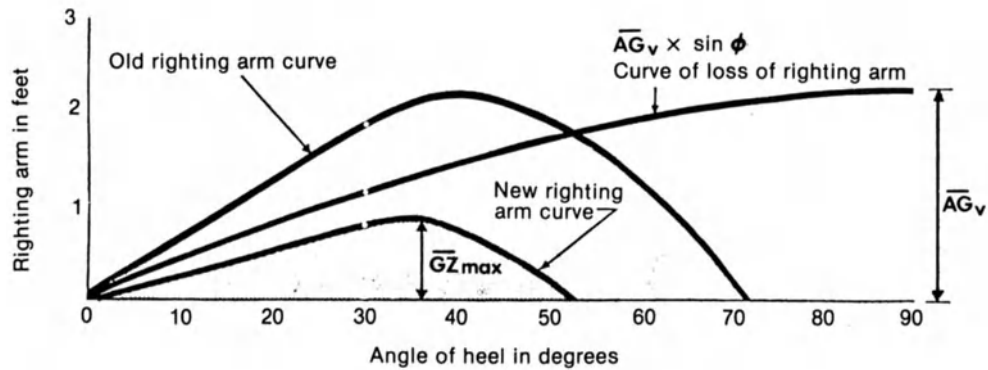


Figure 8-8. Curve of static stability as corrected for a loss of stability due to a vertical weight shift

While the above procedure is a simple way of plotting and obtaining the final corrected curve, it should be kept in mind that one is actually subtracting from or adding to the original stability curve the correction $\overline{AG}_v \sin \phi$ through the range of angles of inclination.

The sine correction is applicable for all vertical movements of the position of the center of gravity from any assumed or uncorrected position, that is, vertical weight shifts and weight additions.

8-4 Transverse Correction for Position of G

The consideration of an athwartship shift in G , while not applicable to the intact, symmetrically loaded ship, nevertheless follows logically any discussion of the vertically corrected stability curve.

The reasoning applied to the vertical correction may be similarly applied to the transverse relocation of G .

Let the actual position of the center of gravity be at some position G (figure 8-9), which is on a line perpendicular to the vertical centerline plane passing through

the original center of gravity A . In other words, it is off center by the amount $\overline{AG}_T$, where the subscript denotes a transverse shift.

From figure 8-9, the correct righting arm is

$$\overline{GZ} = \overline{AZ}_0 - \overline{AR}$$

but

$$\overline{AR} = \overline{AG}_T \cos \phi$$

$$\overline{GZ} = \overline{AZ}_0 - \overline{AG}_T \cos \phi \quad (8-2)$$

where $\overline{AZ}_0$ = the original or uncorrected righting arm based on the assumed center of gravity A .

This means, in effect, that the original righting arm ($\overline{AZ}_0$ in this case) has been reduced, as a result of a horizontal weight shift, by the distance that the center of gravity of the ship has moved transversely from the centerline times the cosine of the angle of inclination.

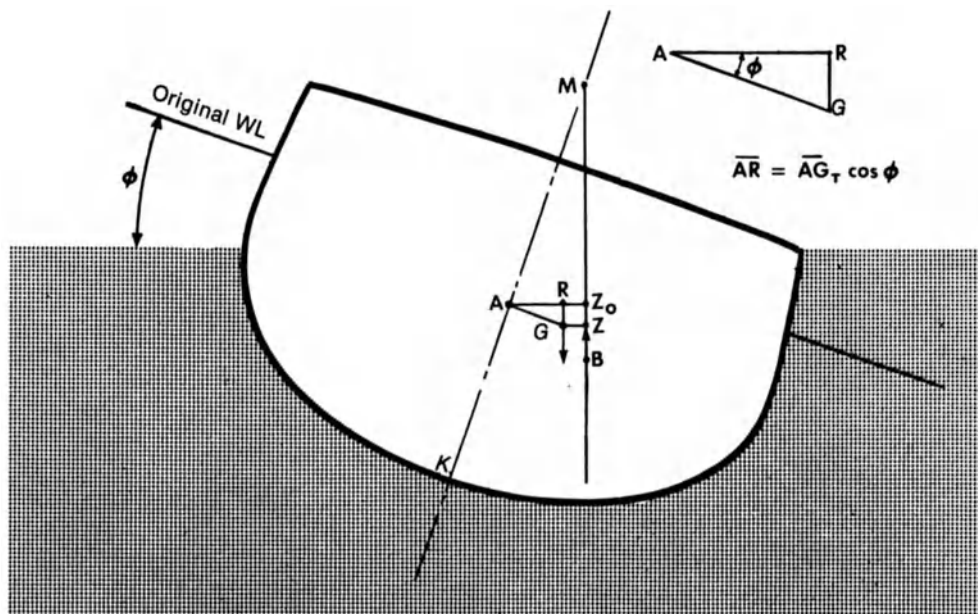


Figure 8-9. Loss of righting arm when the center of gravity is moved off the centerline

To establish equilibrium conditions, any heeling moment existing on the ship must be exactly balanced by a righting moment at some angle of inclination on the side where the heeling moment exists. If the heeling moment does not vary with time, a list results. In addition to the list, the overall stability characteristics suffer a deterioration on the side where the list exists. These two primary effects may be shown most effectively on the plot of static stability in figure 8-10. In this plot, the original curve of static stability (already corrected for any vertical shift) indicates an upright ship with positive righting arms, as shown through a range of 105° . The figure further indicates that a transverse weight shift has resulted in a reduction of the righting-arm curve by $\overline{AG}_T \cos \phi$, plotted as a cosine correction curve.

It should be noted that when the stability curve both to port and starboard is being investigated, this correction curve is conventionally plotted above the axis when G is on the starboard side, and below the axis when G is on the port side. If, on the other hand, the stability curve for only one side is being constructed, it is customarily constructed in the first quadrant. In this case, the cosine correction curve is drawn above the horizontal axis when the ship's center of gravity is off center on that side of the ship for which the stability curve is being constructed. Where the center of gravity is off center to the other side, the cosine correction curve is drawn below the horizontal axis.

Where the cosine correction curve intersects the righting-arm curve, the point of equilibrium is estab-

lished. This also establishes the angle of list at the value ϕ_1 , where the corrected righting arm is zero. The ordinates of the final or corrected overall stability curve are obtained by subtracting the ordinate of the cosine correction curve from the original stability curve. These ordinates are plotted on the axis of the original curve. The reduction in overall stability may be readily seen by comparing the original and corrected stability curve.

The transverse weight shift resulting in the shift of G has done nothing to alter the vertical height of G nor the displacement Δ . Therefore it is correct to say that the vertical distance between G and M has not changed, since the position of M depends upon the displacement. For horizontal weight shifts, the original value of the metacentric height $\overline{GM}$ remains unchanged. The effect on the range of stability must be considered on the basis of the individual changes of the two limits bounding the new, positive-righting-arm curve. The lower limit is at the angle of permanent list, while the upper limit of static stability may be increased or decreased, depending on whether the original range was greater or less than 90° . In figure 8-10, the corrected range to starboard is from about 17° to nearly 115° .

On the side opposite to the angle of list caused by the off-center location of G , the magnitude of the righting arms is correspondingly increased in the amount $\overline{AG}_T \cos \phi$. This is not of prime importance to the damage-control officer, because he is obviously more concerned with the residual stability on the side where it has been reduced and where the list exists. Where the center of gravity is off the centerline and the re-

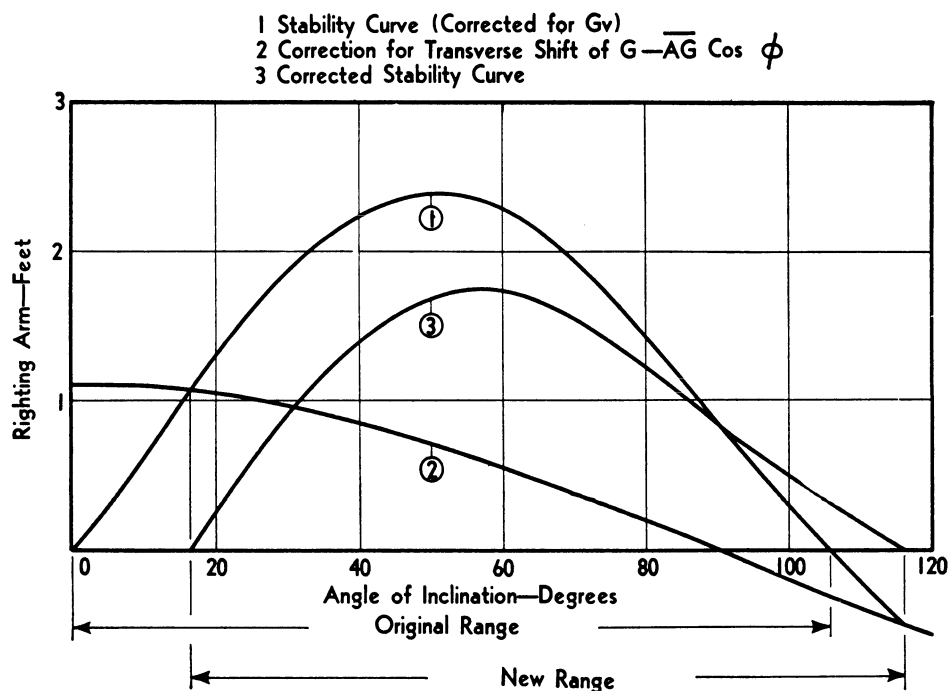


Figure 8-10. Correction to static-stability curve for transverse shift of G to starboard

sulting stability curve is unsymmetrical, a plot of the stability curve for 90° each side of the upright gives a complete picture of the stability situation. Figure 8-11 shows the corrections for both an upward and starboard movement of G leading to such a plot. The algebraic sum of the corrections applied successively to the static-stability curve results in the final stability curve, shown in figure 8-11 and expressed as

$$\overline{GZ}(\phi) = \overline{AZ}_0 - \overline{AG}_V \sin \phi - \overline{AG}_T \cos \phi \quad (8-3)$$

8-5 Metacentric Height and Stability Curves

Recall that

$$\overline{GZ} = \overline{GM} \sin \phi$$

for small ϕ . Also for small ϕ ,

$$\sin \phi \approx \phi \text{ (radians)}$$

Therefore,

$$\overline{GZ} = \overline{GM} \phi \text{ (radians)}$$

or

$$\overline{GM} = \overline{GZ}/\phi \text{ (radians)} \quad (8-4)$$

for small ϕ . Considering the static-stability curve, the expression $\overline{GZ}/\phi$ is the slope of the curve, which means that $\overline{GM}$ equals the slope of the static-stability curve as ϕ approaches zero. The slope of a given static-stability curve can be found by drawing a tangent to the curve, at ϕ equals zero, and measuring the slope in ft/degree. To find $\overline{GM}$, the slope in these units must be multiplied by a conversion factor of $180/\pi$. An alternative graphical method to obtain $\overline{GM}$ is to erect a perpendicular

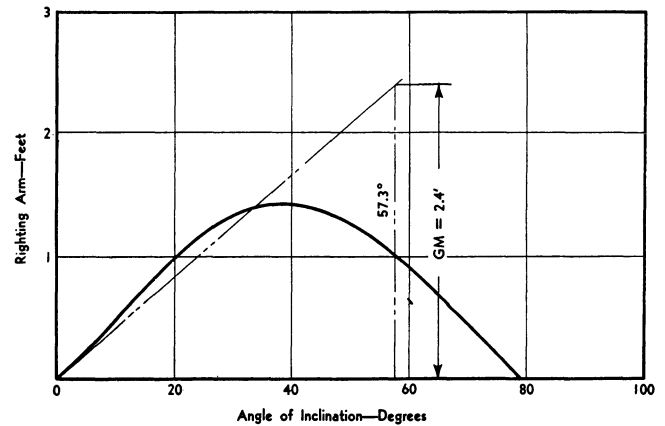


Figure 8-12. Metacentric height

on the stability curve of figure 8-12 at the angle of inclination of 57.3° (one radian) and draw a tangent to the curve at the origin. The height of the intersection of this tangent with the perpendicular will equal the $\overline{GM}$ on the righting-arm scale.

When the ship's center of gravity is not on the centerline, $\overline{GM}$ cannot be determined by measuring the slope of the static-stability curve, which has been corrected for the off-center position of the center of gravity. $\overline{GM}$ remains unchanged due to a horizontal weight shift. Hence, to determine $\overline{GM}$ graphically it is necessary to measure the slope at the origin of the static-stability curve—drawn for the ship's center of gravity on the centerline—but to measure it at its final, vertical height. In other words, in figure 8-10, $\overline{GM}$ must be

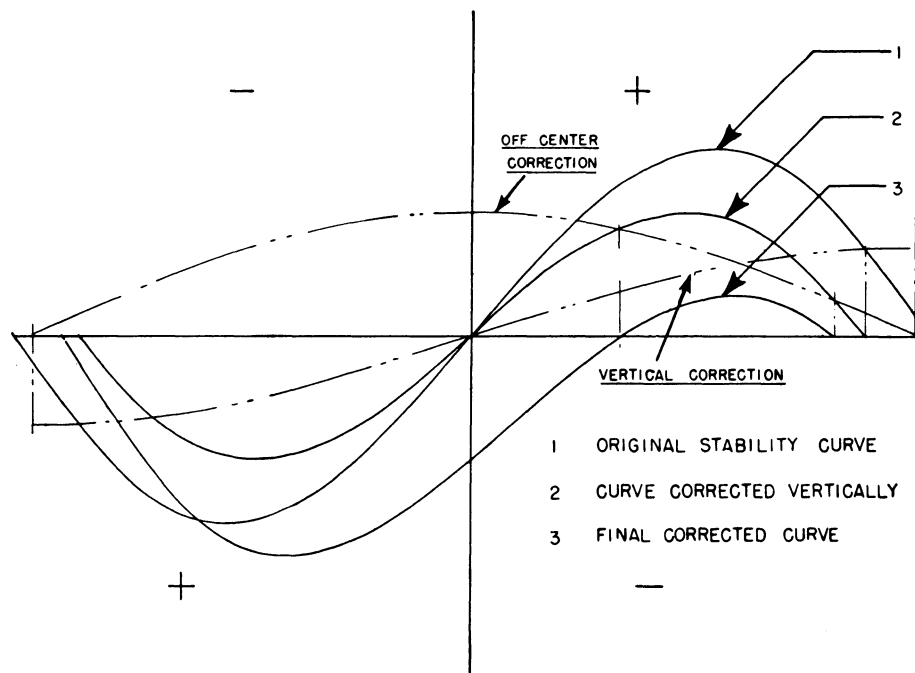


Figure 8-11. Static-stability curve showing vertical and transverse corrections port and starboard

determined by the slope at the origin of curve one, corrected for G_v , rather than by the slope of curve 3, the corrected stability curve.

The variation of $\overline{GM}$ with displacement and vertical location of G may be approximated from the cross curves of stability (without constructing a static-stability curve) by differentiating equations 8-1 and 6-1 with respect to ϕ .

$$\begin{aligned}\overline{GZ}(\phi) &= \overline{GM} \sin \phi \\ &= \overline{AZ}(\phi) - \overline{AG}_v \sin \phi \\ \frac{d\overline{GZ}(\phi)}{d\phi} &= \overline{GM} \cos \phi \\ &= \frac{d\overline{AZ}(\phi)}{d\phi} - \overline{AG}_v \cos \phi \\ \overline{GM} &= \frac{1}{\cos \phi} \frac{d\overline{AZ}(\phi)}{d\phi} - \overline{AG}_v \quad (8-5)\end{aligned}$$

The rate of change of $\overline{AZ}(\phi)$ with ϕ may be approximated by the value of $\overline{AZ}$, at ϕ equal to 10° , read from the cross curves of stability divided by 10° expressed in radians. The error in using the 10° value of $\overline{AZ}$, which is the first one shown on the cross curves, is small, since $\cos(10^\circ)$ is close to unity.

$$\overline{GM} \approx \frac{\overline{AZ}(10^\circ)}{10 \frac{\pi}{180}} - \overline{AG}_v \quad (8-6)$$

Example 8-1: Compare $\overline{GM}$ calculated by equation 3-12 with the value approximated by equation 8-5 for a DD 692 destroyer displacing 3,600 tons with a $\overline{KG}$ of 15.5 feet.

Solution: From the curves of form,

$$\begin{aligned}\overline{KM} &= 19.0 \text{ ft} \\ \overline{GM} &= \overline{KM} - \overline{KG} = 19.0 - 15.5 \\ &= \boxed{3.5 \text{ feet}}\end{aligned}$$

From equation 8-6,

$$\overline{GM} = \frac{\overline{AZ}(10^\circ)57.3}{10} - \overline{AG}_v$$

From figure 8-3,

$$\begin{aligned}\overline{AZ}(10^\circ) &= 0.89 \text{ ft} \\ \overline{GM} &= \frac{0.89(57.3)}{10} - (15.5 - 14) \\ &= 5.10 - 1.5 = \boxed{3.6 \text{ feet}}\end{aligned}$$

which compares favorably.

The $\overline{AZ}(10^\circ)$ value from the cross curves is difficult to estimate more accurately than ± 0.02 feet, which is only ± 0.11 feet for $\overline{GM}$, so the method approaches the

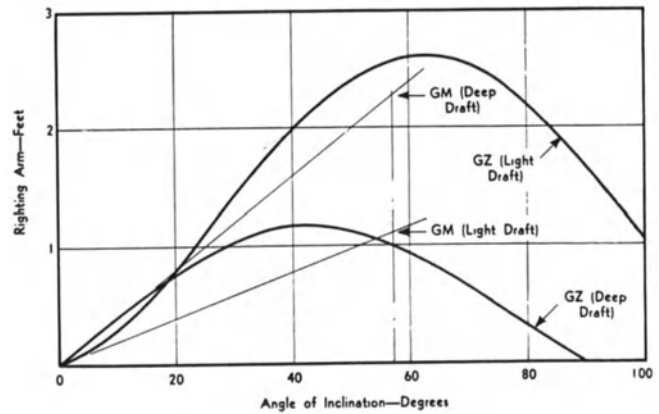


Figure 8-13. Comparison of stability curves of a ship in two extreme displacement conditions (same $\overline{KG}$)

accuracy of the technique based on a tangent to the static-stability curve. The latter method is simpler and quicker if the corrected static-stability curves already exist.

8-6 Common Stability Characteristics

It should be emphasized that there is little relationship between $\overline{GM}$ and $\overline{GZ}$ except at small ϕ . However, there is a common characteristic for certain vessels of destroyer-type form and larger vessels with fine ends. This similarity is such that there is a consistent variation of $\overline{GM}$ and $\overline{GZ}$ where the magnitude of one varies inversely with the magnitude of the other between deep and light draft. This characteristic is illustrated in figure 8-13.

Many ships have this characteristic of a gentle initial slope in the stability curve, which indicates a moderate or low $\overline{GM}$ value. The slope of this curve increases sharply as the angle increases.

If such a curve is related to a case where the center of gravity is corrected for a higher position, it can be seen from figure 8-14 that this initial "dip" in the curve may create a condition where the $\overline{GM}$ becomes negative. Such a condition will result in a symmetrical list

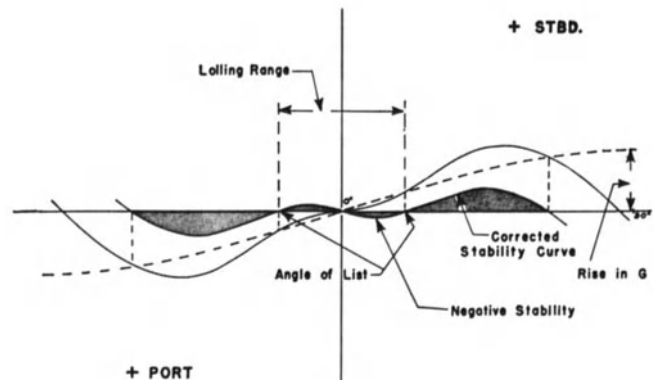


Figure 8-14. Static-stability curve showing range of instability (lolling)

as shown. In this condition, the ship will not remain upright but will list to one side or the other, depending on the direction of any initial inclining force, until the changing underwater shape restores positive stability at the angle of list. This situation is generally described as *lolling*. The most common example of lolling is an unloaded cargo vessel where, in the light condition, the center of gravity has risen to a level at which the sine correction curve intersects the stability curve at a low angle of heel and is greater than the righting-arm curve at lesser angles, as shown in figure 8-14. In a cargo vessel, this situation is not serious because the stability characteristics are designed for a wide range of vertical locations of center of gravity, and even though the $\overline{GM}$ may be negative, there will be sufficient positive stability to maintain a satisfactory range and maximum righting arm. Furthermore, this condition can readily be corrected by reloading or ballasting. However, no naval vessels are so designed, and any lolling indicative of a negative $\overline{GM}$ should be viewed with considerable concern.

In summary, a ship can assume a list in two fundamental situations: when an off-center weight moves the center of gravity out of the vertical centerline plane, causing an inclining moment; and when a high center of gravity exists in ships whose corrected stability curves show a negative righting arm in the initial stages of inclination. The former case will, of course, result in an overall unsymmetrical condition of stability and a list on the side toward which the center of gravity moves. The latter case will result in a list on either side with an overall symmetrical condition of stability.

8-7 General-Stability Diagram

In preceding discussions in this text, the ship's static-stability curves treated—as a development from the cross curves of stability—the various applications of corrections for vertical and transverse shifts of centers of gravity. Generally these problems were handled separately, working from the cross curves and with the various sine and cosine correction curves. This process can often be laborious. It can also be particularly awkward in time of action, in anticipation of or under duress of battle in a damaged condition. Consequently, a more rapid means of resolving the situation must be adopted.

One successful method, which can be used to simplify the work in obtaining the final corrected stability curve, is derived from what is shown as the *general stability diagram*. This diagram, although not the basis for a standard form aboard every ship, can be easily prepared from the cross curves of stability. A typical example for the DD 692 destroyer is provided in figure 8-15.

The diagram consists of a grid divided into three parts. The upper part contains a family of uncorrected righting-arm (static-stability) curves lifted from the cross

curves of stability for a variety of displacements (spaced at regular intervals). On top of this family of righting-arm curves, which should cover anticipated operating conditions, is superimposed a family of sine curves for regular intervals of vertical shifts in the center of gravity. The cross curves of the DD 692, from which the uncorrected static-stability curves were taken, were constructed for the assumed $\overline{KG}$ of 14 feet.

In the middle part of the grid, there is a family of cosine curves for regular intervals of off-center shifts of center of gravity.

The lower part of the grid is left blank for the plotting of the final curve of stability, corrected, from the upper two parts.

To use this general stability diagram properly the following steps are suggested:

1. Select the stability curve in the upper group that corresponds to the displacement of the ship for which the resultant curve is desired. If the corrected condition involves the addition of a weight, such as in cases of flooding, the displacement selected must include this added weight. It is unlikely that the actual displacement will coincide with one of the printed curves. It will be necessary, then, to construct a curve in pencil between two printed curves, estimating the percentage difference between the curves for the actual displacement used.

2. Select a sine curve on the upper part of the grid whose maximum ordinate (at 90°) corresponds with the final $\overline{KG}$ for which the stability curve is to be corrected. Here again, the effective $\overline{KG}$ value will probably not be one of the printed curves, but rather some value in between two constructed values. As in the case of the stability curve, draw a curve in pencil, interpolating between the two curves throughout the range. Note that the ordinate value for these sine curves is the correction to the righting arm for the vertical movement of the center of gravity from the assumed position (from which the cross curves were drawn) to the final position as calculated.

3. Transfer the residual righting arms to the middle part of the grid and plot the new righting-arm curve (that is, the difference between the uncorrected stability curve and the sine correction). (If there are no off-center weights, go directly from the upper grid to the lower grid.) This operation is accomplished in a manner similar to that used for replotting any corrected stability curve. In this case, the difference in ordinates between the two selected curves is simply moved down the vertical grid lines for every 10° angle of inclination and plotted on the middle grid axes.

4. Select a cosine curve in the middle part of the grid that corresponds at the maximum ordinate (at 0°) to the athwartship distance of G from the centerline. Here

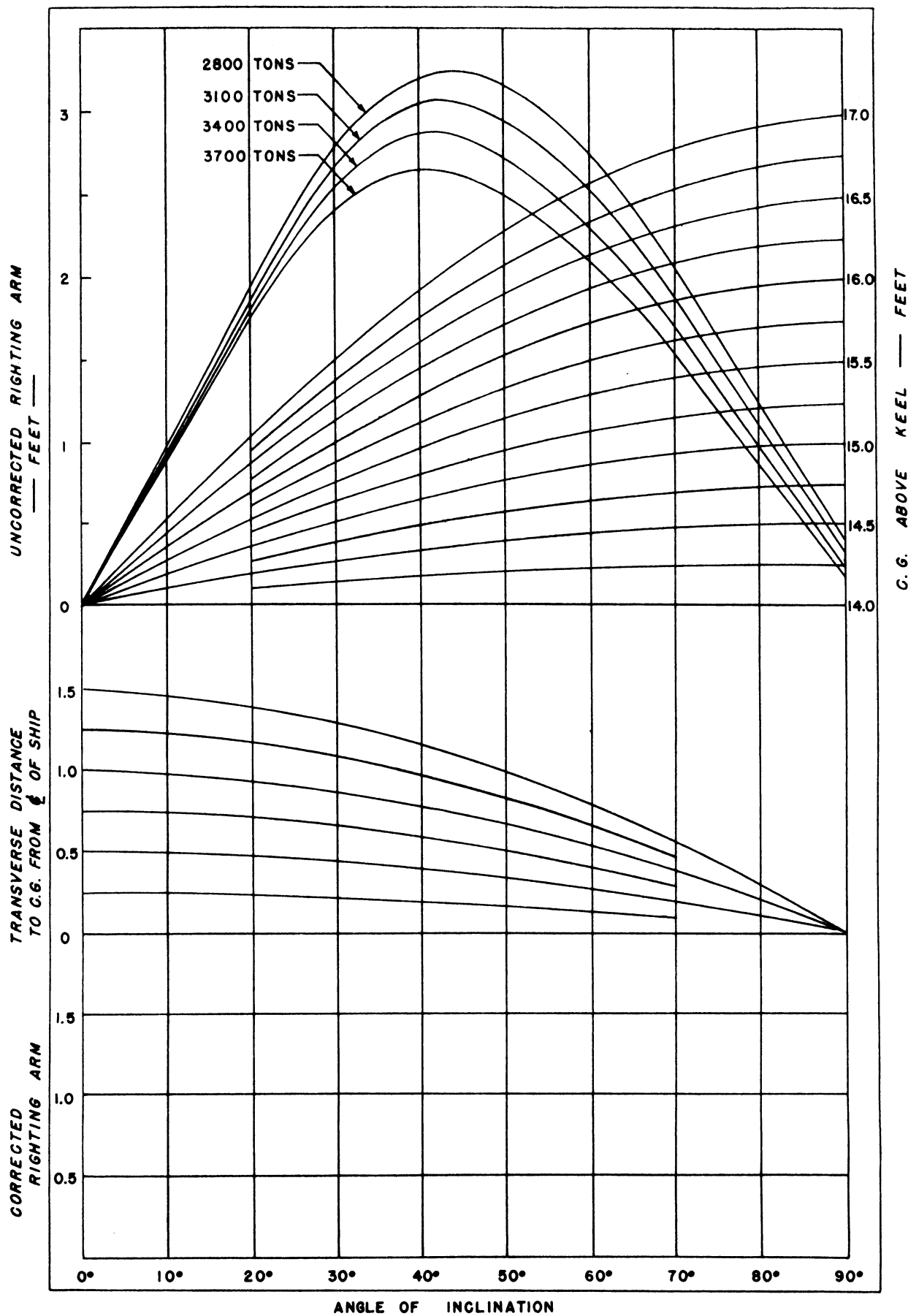


Figure 8-15. General-stability diagram for a DD 692 (long hull)

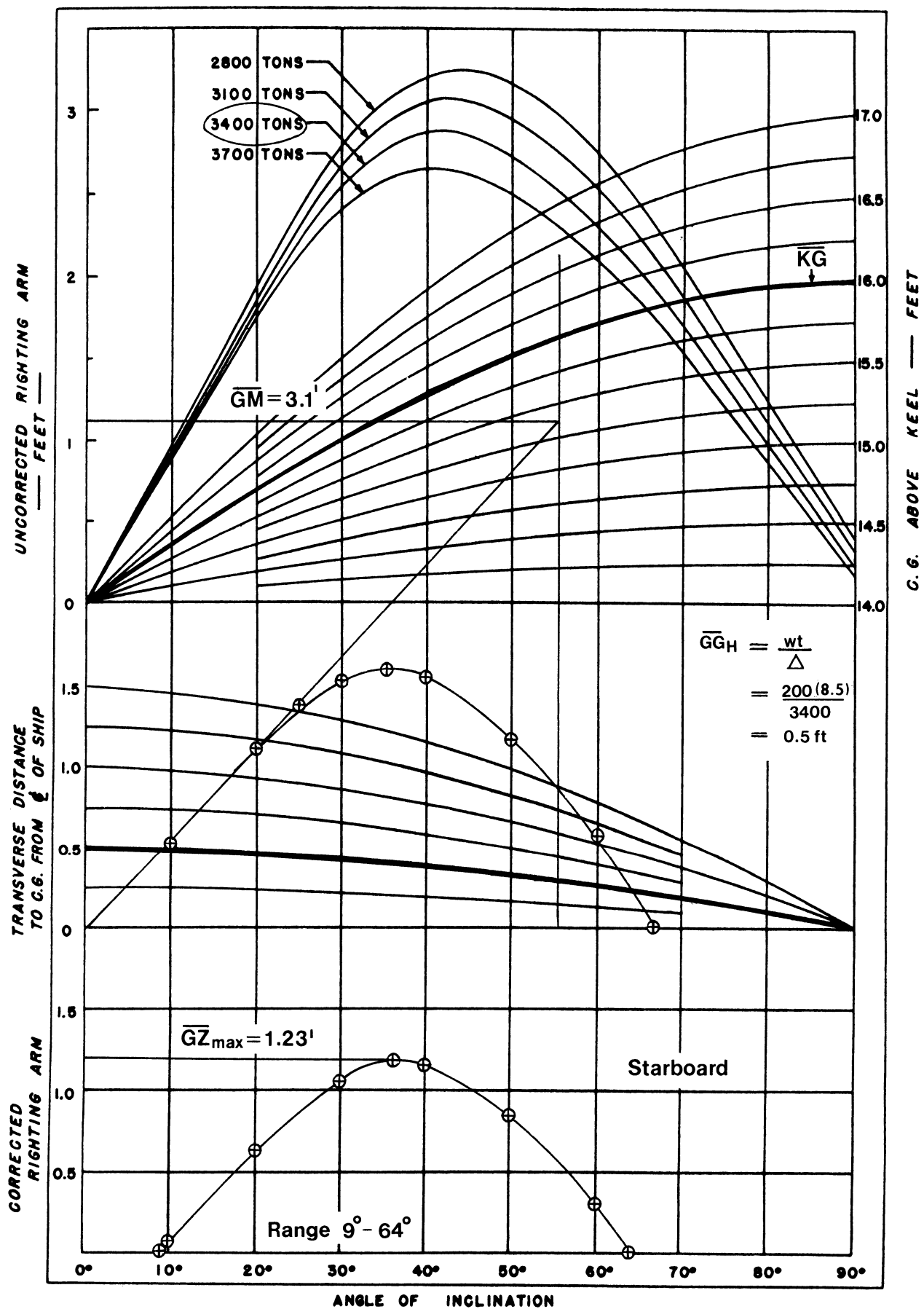


Figure 8-16. General-stability diagram for a DD 692 (long hull)

again, the exact value will probably fall between two printed curves, and the actual correcting cosine curve will have to be traced in with pencil.

5. Transfer the final righting arms (the difference between the step 3 and step 4 curves) from the middle grid to the lower grid. When this is done note the two angles that establish the range of stability. These two points must correspond with the intersections of the righting-arm curve and the cosine curve established in the middle grid.

It should be noted that if the angle of list is known and the off-center position of G is not known, it is possible to locate the proper cosine correction curve by spotting the point on the righting-arm curve at the angle of list. Thus, this point, which must be on the cosine curve desired, establishes the proper correction.

Example 8-2: A DD 692-class destroyer operating off Greenland experiences severe icing topsides. Two hundred tons of ice are distributed more heavily on the windward side (starboard) than on the leeward side. Assume the center of gravity of the ice is 32 feet above the keel, 40 feet forward of midships, and 8.5 feet to starboard of the centerline of the ship. Before the icing occurred, the ship had a displacement of 3,200 tons and a $\overline{KG}$ of 15 feet with level trim.

Analyze the range of stability to starboard on a general stability diagram and determine the list angle and the maximum righting arm for the icing condition. Estimate $\overline{GM}$ and ϕ_{List} by two methods.

Solution: From the curves of form at

$$\Delta_1 = 3,400 \text{ tons} = \Delta_0 + 200 \text{ tons of ice,}$$

$$T_M = 14.2 \text{ ft}$$

$$\overline{KM} = 955 \left(\frac{2}{100} \right) = 19.1 \text{ ft}$$

$$\begin{aligned} \overline{KG}_1 &= \frac{\Delta_0 \overline{KG}_0 + w \overline{Kg}}{\Delta_1} \\ &= \frac{3,200(15) + 200(32)}{3,400} \end{aligned}$$

$$= 16.0 \text{ ft}$$

$$\therefore \overline{AG}_v = 2 \text{ ft}$$

$$\overline{GM} = \overline{KM} - \overline{KG} = 19.1 - 16 = \boxed{3.1 \text{ feet}}$$

$$\tan \phi = \frac{wt}{\Delta_1 \overline{GM}_1} = \frac{200(8.5)}{3,400(3.1)}$$

$$= 0.161$$

$$\phi_{\text{List}} = \boxed{9.2^\circ}$$

$$\overline{GG}_T = \frac{wt}{\Delta_1} = \frac{200(8.5)}{3,400}$$

$$= 0.5 \text{ ft}$$

Figure 8-16 illustrates the application of steps 1 through 5 to the general stability diagram for this problem. The analytical and graphical estimates of $\overline{GM}$ agree, as do the estimates of the list angle.

8-8 Free-Surface Correction at Large Angles of Heel

In chapter 6, section 6-7, the free-surface correction to the righting arm $\overline{GZ}$ for moderate angles of heel was given by equation 6-31.

$$\overline{G_1 Z_1} = \left[\overline{GM} - \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla_s} \left(1 + \frac{\tan^2 \phi}{2} \right) \right] \sin \phi$$

The limits of applicability for this equation were for a fixed position of the metacentric height M and for the liquid-free surface not coming into contact with the top of the tank (pocketing). Before applying a free-surface correction to the static-stability curves, however, an analytical solution for “wall-sided ships” up to the point of deck-edge submergence will be developed.

At angles of heel greater than about 10° , there is a significant shift in the vertical location of the center of buoyancy. This is because at all angles of inclination the location of the centroid of the immersed wedge is further from the keel than the centroid of the emerged wedge, as illustrated in figure 8-17. Beyond 10° of inclination the shift becomes significant. By analogy with the development of the shifts in the center of gravity caused by a free surface (equations 6-29 and 6-30), the horizontal and vertical shifts in the center of buoyancy are

$$\overline{BB}_{1_T} = \overline{BM} \tan \phi = \frac{I}{\nabla} \tan \phi \quad (8-7)$$

$$\overline{BB}_{1_V} = \overline{BB}_{1_T} \frac{\tan \phi}{2} = \overline{BM} \frac{\tan^2 \phi}{2} \quad (8-8)$$

Note that the vertical shift in the center of buoyancy increases the height of the metacenter and also increases the righting arm.

The effect on $\overline{GZ}$, along with the effect of free surface, is illustrated in figure 8-18. The resulting $\overline{G_1 Z_1}$ is given by

$$\begin{aligned} \overline{G_1 Z_1} &= \overline{GZ} + \overline{BB}_{1_V} \sin \phi \\ &\quad - \overline{GG}_{1_T} \cos \phi - \overline{GG}_{1_V} \sin \phi \\ &= \overline{GM} \sin \phi + \left(\overline{BM} \frac{\tan^2 \phi}{2} \right) \sin \phi \\ &\quad - \left(\frac{\gamma_t}{\gamma_s} \frac{i}{\nabla} \tan \phi \right) \cos \phi \\ &\quad - \left(\frac{\gamma_t}{\gamma_s} \frac{i}{\nabla} \frac{\tan^2 \phi}{2} \right) \sin \phi \\ \overline{G_1 Z_1} &= \left[\overline{GM} + \overline{BM} \frac{\tan^2 \phi}{2} \right. \\ &\quad \left. - FSC \left(1 + \frac{\tan^2 \phi}{2} \right) \right] \sin \phi \quad (8-9) \end{aligned}$$

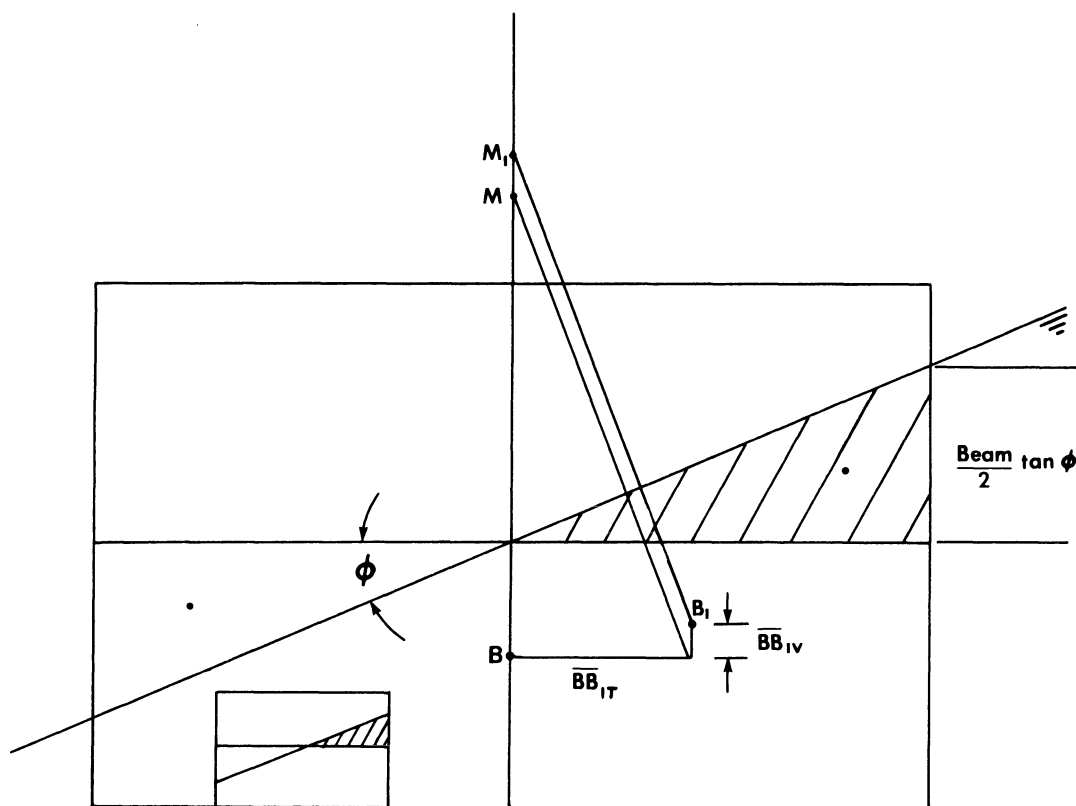


Figure 8-17. Variation of $\overline{BM}$ for a wall-sided ship

This equation is valid for wall-sided ships with wall-sided tanks up to the point of deck-edge submergence or pocketing, whichever comes first.

POCKETING

The effect of free surface is always modified to some extent by pocketing. Such a modifying effect occurs when liquid comes into contact with the top of the tank or the bottom surface of the tank is exposed. Either of these situations generally takes place at some finite inclination, pockets the liquid in one side of the tank, and reduces the breadth of the free-surface area. Depending on the size and shape of the tank and the amount of liquid it contains, this becomes an extremely variable factor and one not subject to a simple solution (Comstock 1967, chapter 2, section 5). However, because pocketing always reduces the increasing area of free surface, its effect is to prohibit the realization of unrestricted free surface throughout the complete range of stability. In the case of deep, narrow tanks, the effect of the increasing free-surface area and the ultimate effect of pocketing is one of modifying the unrestricted effect of increased free-surface area. Hence, if it is ne-

glected, pocketing may be considered an indeterminate factor of safety, since the full free-surface effect describes extreme conditions.

The practice of carrying fuel oil tanks 95 percent full instead of pressed full to allow for thermal expansion of the oil, results in a free surface that pockets at small angles of inclination. Although $\overline{GM}$ and initial righting arms may be slightly reduced by the free-surface effect, the pocketing prevents an appreciable loss of overall stability.

Example 8-3: A petroleum-products carrier is 700 feet long, 120 feet wide, and has a molded depth of 75 feet. It has a C_B of 0.9, C_{WP} of 0.9, C_{IT} of 0.81, and can be considered a wall-sided ship up to 30° of heel when loaded to a draft of 40 feet. It is divided into eighteen cargo tanks 35 feet wide by 90 feet long, arranged in a 3-by-6 matrix. Estimate $\overline{GM}$ and the righting-arm curve in 5° increments up to 30° for a $\overline{KG}$ of 41 feet,

- assuming the cargo tanks are pressed full of petroleum ($\gamma_t = 0.9$)
- assuming a free surface without pocketing up to an angle of 30° in all eighteen tanks

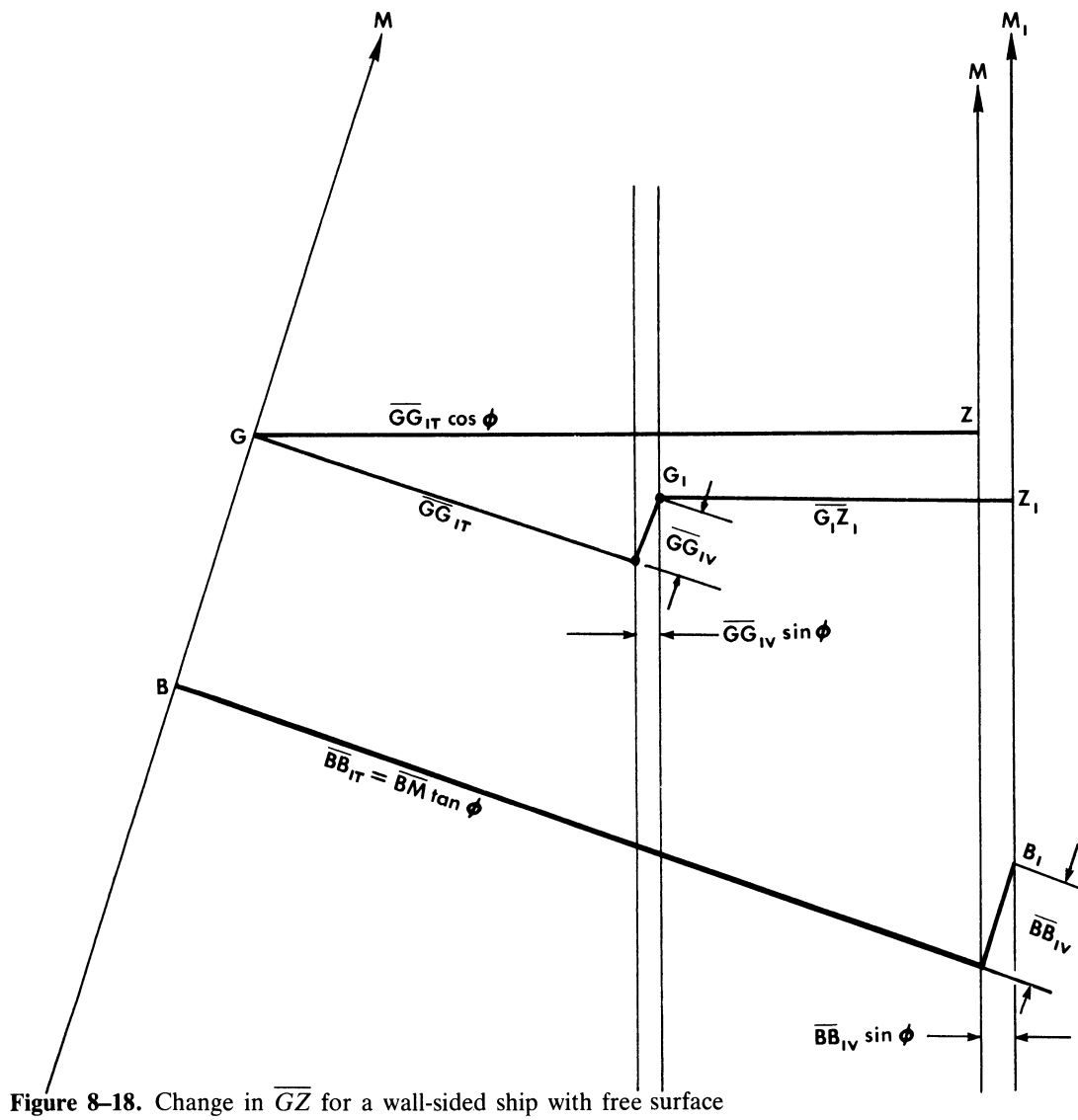


Figure 8-18. Change in $\overline{GZ}$ for a wall-sided ship with free surface

Solution: From equations 3-5 and 3-10,

$$\begin{aligned}\overline{KB} &= T \left(\frac{C_{WP}}{C_B + C_{WP}} \right) = 40 \left(\frac{0.9}{0.9 + 0.9} \right) \\ &= 20 \text{ ft}\end{aligned}$$

$$\overline{BM} = \frac{C_{IT} B^2}{C_B 12T} = \frac{0.81}{0.9} \frac{120^2}{12(40)}$$

$$= 27 \text{ ft}$$

$$\begin{aligned}\overline{GM} &= \overline{KB} + \overline{BM} - \overline{KG} \\ &= 20 + 27 - 41 \\ &= \boxed{6 \text{ feet}}\end{aligned}$$

a) $\phi \leq 30^\circ$

$$\begin{aligned}\overline{G_1Z_1} &= \left[\overline{GM} + \overline{BM} \frac{\tan^2 \phi}{2} \right] \sin \phi \\ &= \left[6 + 27 \frac{\tan^2 \phi}{2} \right] \sin \phi\end{aligned}$$

b) $\phi \leq 30^\circ$

$$\begin{aligned}FSC &= \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla_s} \times 18 \text{ tanks} \\ &= \frac{0.9}{1.025} \frac{35^3(90)18}{12(.9)(700)(120)(40)} \\ &= 1.68 \text{ ft}\end{aligned}$$

$$\overline{G_1Z_1}(\text{eff}) = \overline{G_1Z_1} - FSC \left(1 + \frac{\tan^2 \phi}{2} \right) \sin \phi$$

Thus,

$$\begin{aligned}\overline{GM}_{\text{eff}} &= \overline{GM} - FSC \\ &= 6 - 1.68 \\ &= \boxed{4.3 \text{ feet}}\end{aligned}$$

The analysis is set up in table 8-1 and plotted in figure 8-19. The uncorrected $\overline{GZ}$ curve shows an increasing slope of the righting arm with an increasing angle of heel, which is characteristic of wall-sided ships.

The situation of having a free surface in all eighteen tanks is not realistic if good cargo-handling practices are followed. Conceivably, a free-surface correction of this magnitude could occur at an open-sea mooring during loading operations, if the tanks were not completely filled, one at a time, to avoid severe bending-moment problems. A proper filling sequence could prevent this.

Although it is tempting to treat the free-surface correction as a sine correction, table 8-2 demonstrates that the correction is actually closer to a tangent correction up to heel angles of 45° , without pocketing. If the functions of ϕ are expanded using trigonometric identities, the correction is close to $\tan \phi$ correction, even for small angles, where

$$\begin{aligned}\frac{\tan^2 \phi}{2} &\approx 2 \sin^2 \frac{\phi}{2} \\ \left(1 + \frac{\tan^2 \phi}{2} \right) \sin \phi &= \left(1 + \frac{\tan^2 \phi}{2} \right) \tan \phi \cos \phi \\ &= \tan \phi \left[1 + \frac{\tan^2 \phi}{2} \right. \\ &\quad \left. - 2 \sin^2 \frac{\phi}{2} \right. \\ &\quad \left. - 2 \frac{\tan^2 \phi}{2} \sin^2 \frac{\phi}{2} \right] \quad (8-10)\end{aligned}$$

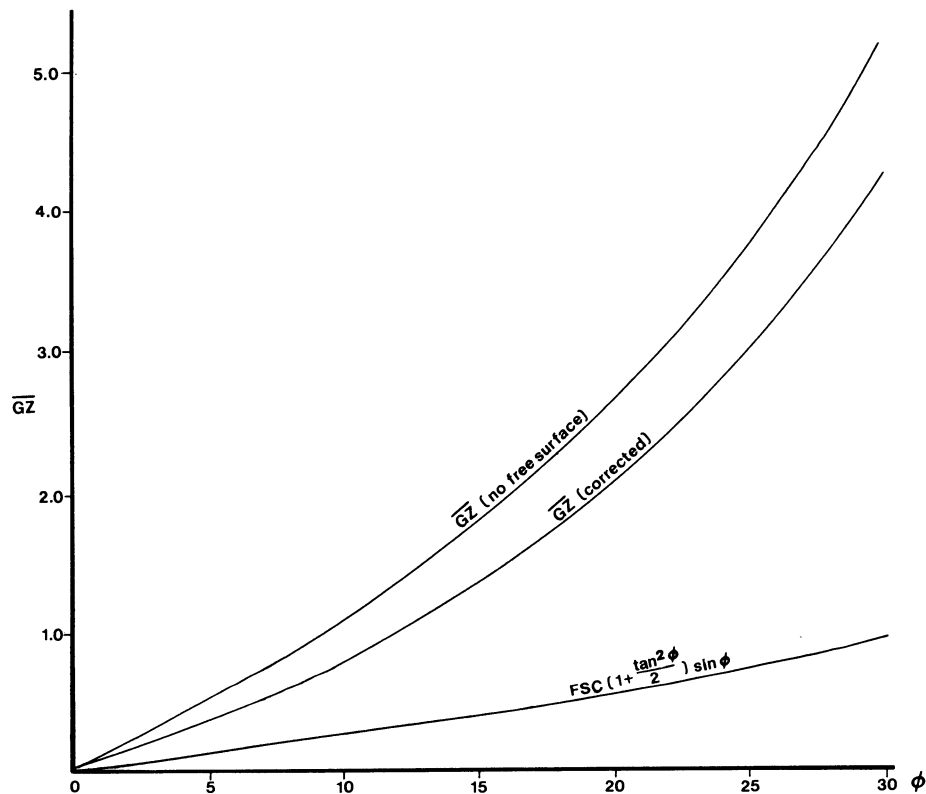


Figure 8-19. Righting arm for a wall-sided ship

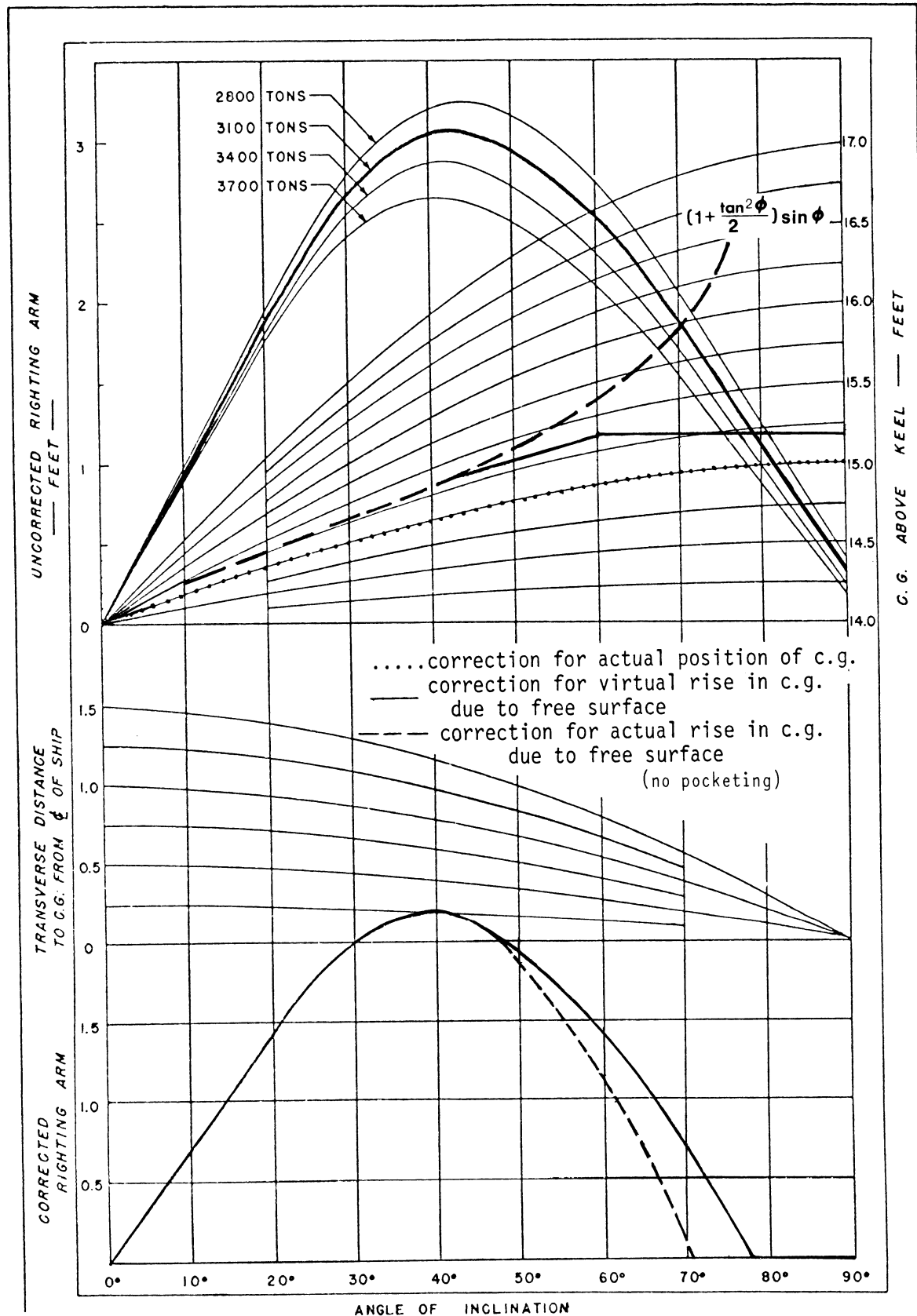


Figure 8-20. General-stability diagram for a DD 692 (long hull)

Table 8-1.

ϕ	$\overline{GM} + \overline{BM} \frac{\tan^2 \phi}{2}$ ft	$\overline{GZ}$ ft	$FSC \left(1 + \frac{\tan^2 \phi}{2} \right) \times \sin \phi$ ft	$\overline{GZ}_{corr}$ ft
0	6	0	0	0
5	6.10	0.53	0.15	0.38
10	6.42	1.11	0.30	0.81
15	6.97	1.80	0.45	1.35
20	7.79	2.66	0.61	2.05
25	8.94	3.78	0.79	2.99
30	10.5	5.25	0.98	4.27

Table 8-2. Free-Surface Correction at Large Angles of Heel

$$= \frac{\rho_t}{\rho_s} \frac{i}{\nabla} \left(1 + \frac{\tan^2 \phi}{2} \right) \times \sin \phi$$

ϕ	$\sin \phi$	$\tan \phi$	$1 + \frac{\tan^2 \phi}{2}$	$\left(1 + \frac{\tan^2 \phi}{2} \right) \times \sin \phi$
5	.0872	.0875	1.0038	0.0875
10	.1736	.1763	1.0155	0.1763
15	.2588	.2680	1.0359	0.2681
20	.3420	.3640	1.0662	0.3647
25	.4226	.4663	1.1087	0.4686
30	.5000	.5774	1.1667	0.5833
35	.5736	.7002	1.2452	0.7142
40	.6428	.8391	1.3520	0.8691
45	.7071	1.000	1.5000	1.0607

Since pocketing is likely to occur for heel angles greater than 45°, a modified procedure can be used, as illustrated in figure 8-20. The sequence is as follows:

1. Calculate the $FSC \left(\text{equal to } \frac{\gamma_t}{\gamma_s} \frac{i}{\nabla} \right)$ and add it to the value of the $\overline{AG}_v$ correction on the general stability diagram.
2. Sketch a horizontal line over to the 60° of heel line.
3. Sketch in the sine correction from that point back to zero. Note that the line will intersect the actual correction, without pocketing, at about 45°.

This method gives a conservative estimate of the free-surface correction without being overly pessimistic at large angles of heel.

8-9 Stability Criteria

In chapter 6, there was some discussion of general-stability characteristics relating to metacentric height and the righting-arm curve. It should be properly pointed out that the primary criterion for the safety of a ship is neither the $\overline{GM}$ nor the extent of the righting-arm curve but rather the maximum righting arm and the angle at which it occurs.

Over the years, a number of rather arbitrary criteria have been set forth regarding quantitative aspects of righting arm versus heeling arm by ship-design agencies. One particular study worth citing is a part of the stability-design procedure as well as the operational criteria followed by the U.S. Navy (Sarchin and Goldberg 1962). For the intact ship these heeling arms originate from external or operational sources, such as strong wind or centrifugal force in a high-speed turn.

Beam winds and *rolling* often combine, creating a hazardous situation. A ship heeled in still water by an external force requires only sufficient righting moment to overcome the heeling moment. However, when rolling action caused by wave motion is introduced, an additional allowance of dynamic stability is required to absorb the energy imparted to the ship by the rolling action. In computing the allowable heeling moments, wind velocities from 50 to 100 knots are used, depending on the type and size of the ship. For vessels intended for all weather operations, 100-knot winds are used.

The heeling arm produced by the wind is calculated from the equation

$$M_w = \frac{.0035 V_w^2 A h \cos^2 \phi}{2,240 \Delta}$$

where V_w = wind velocity

h = height of the ship's lateral resistance above 1/2 draft

A = projected lateral area above $\overline{WL}$

ϕ = angle of heel

Δ = ship displacement.

After plotting the wind's heeling-arm curve (B in figure 8-21), superimposed on the ship's static-stability curve for the existing condition of displacement and $\overline{KG}$, the arbitrary criteria for adequate stability under these circumstances must show that (a) the righting arm at the angle of heel (C in figure 8-21) is not greater than 60 percent of the maximum righting arm, and (b) the area of residual righting energy or dynamic stability (A_1 in figure 8-22) is not less than 140 percent of area A_2 , where this area represents the ship's rolling energy during a 25° roll to windward of point C .

Stability in a turn may sometimes be critical (see chapter 14), particularly in high ships making high-speed turns. The heeling moment produced under such circumstances is the result of the centrifugal force. This expression, discussed in chapter 14, is stated as

$$\frac{\Delta v^2}{gr} \overline{GL} \cos \phi$$

and therefore the heeling arm is

$$\frac{v^2 \overline{GL}}{gr} \cos \phi$$

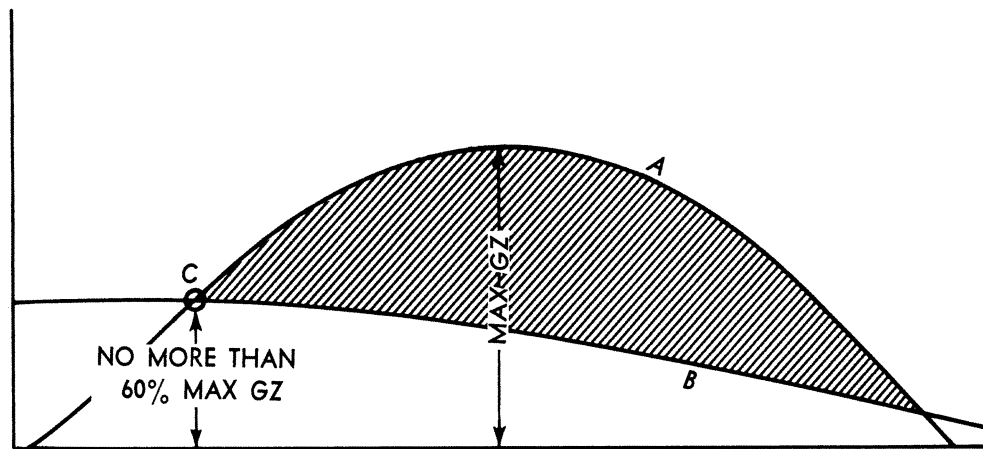


Figure 8-21. Operational stability limitations

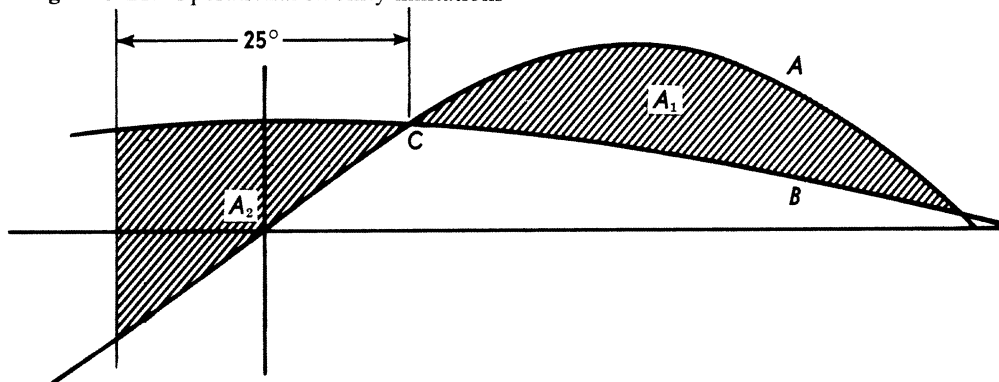


Figure 8-22. Operational stability limitation

where v = ship velocity
 $\overline{GL}$ = distance between the center of gravity and the underwater center of lateral resistance
 g = gravity acceleration
 r = radius of the turning circle.

For practical purposes, r may be considered as half of the tactical diameter.

Under these circumstances, the criterion for adequate stability is again based on the superposition of the heeling-arm curve on the applicable static-stability curve. The stability is considered satisfactory if

1. The angle of steady heel (C in figure 8-22) does not exceed 10° in the case of a new design or 15° for ships in service
2. The heeling arm at point C is not more than 60 percent of the maximum righting arm
3. The residual righting energy or dynamic stability (shaded area in figure 8-22) is not less than 40 percent of the total area (intact or upright dynamic stability) under the ship's stability curve.

It is interesting to note that these same criteria are used for conditions on passenger vessels when a heeling moment develops, because a crowd of passengers moves to one side of the ship.

Problems

8-1. A ship alteration has been proposed for the DD 692-class destroyers to add an antisubmarine helicopter hanger and helo to the superstructure area of the ship. The original conditions before the proposed change are a displacement of 3,200 tons and a $\overline{KG}$ equal to 15 feet. The added structure and helo would weigh 200 tons with a center of gravity 32 feet above the keel. Show all calculations and determine the following, using the general-stability diagram. Use the middle section for (a) and the lower section for (b).

- a) Determine the range of stability, the maximum righting arm, the angle for the maximum righting arm, and the $\overline{GM}$ for the new condition of loading. (Calculate the $\overline{GM}$ by two methods.)
- b) The oil king inadvertently transfers 50 tons of fuel a distance of 17 feet from port to starboard. Determine the angle of list and show the final-stability diagram with the same criteria as in (a).

8-2. A DD 692-class destroyer has a displacement of 3,500 tons and a $\overline{KG}$ of 15.1 feet. Three hundred tons of fuel are removed from a fuel tank whose $\overline{KG}$ is 5.0 feet and whose center of gravity is 4.0 feet to star-

board of the centerline. Complete a general-stability diagram of these conditions and determine the new $\overline{GM}$, the angle of list, the range of stability to port, and the maximum righting arm to port.

8-3. A DD 692-class destroyer operating off Alaska experiences severe icing topsides. It is estimated that 170 tons of ice are distributed more heavily on the windward side (port) than on the leeward side. The center of gravity of the ice appears to be about 25 feet above the keel and the ship heels 10° to port. Before the icing occurred, the ship had a mean draft of 13 feet 6 inches with level trim and a $\overline{KG}$ of 15 feet. Determine the new $\overline{KG}$ and complete a general stability diagram to determine the location of the transverse center of gravity of the ice.

8-4. A 120-ton weight is lifted from a position 5 feet above the keel of a DD 692 (long hull) class ship to 25 feet above the keel and 10 feet to starboard. The DD's displacement is 3,200 tons and $\overline{KG}$ before the weight shift is 15 feet. Determine the following using the general-stability diagram:

- a) final $\overline{GM}$
- b) range of stability (starboard only)
- c) maximum righting arm (starboard)
- d) angle of maximum righting arm

8-5. The USS *Gearing* (DD 710) is at a draft of 13 feet 4 inches forward and 14 feet 2 inches aft in salt water and has a $\overline{KG}$ of 15.5 feet. Due to a leaking stuffing tube the starboard shaft alley floods with 30 tons of salt water. The center of gravity of the flooding

water is midway between stations 15 and 16, 7 feet above the keel and 15 feet to starboard of ship centerline. Ten tons of lube oil is removed from the lube-oil settling tank located in compartment B-2. The oil is 8 feet above the keel, 12 feet to starboard, and 10 feet forward of station 10. In addition feed and fresh water tanks B-8W, B-10W are emptied for repair. The tanks hold a total together of 24 tons of water and have a $\overline{KG}$ of 6 feet, and the center of gravity of the water is at station 7, 20 feet port of centerline.

Determine

- a) initial displacement
- b) new displacement
- c) new mean draft
- d) new $\overline{KG}$
- e) new $\overline{GM}$
- f) angle of list
- g) trimming moment
- h) final trim
- i) final drafts forward and aft

Draw the general stability diagram showing maximum $\overline{GZ}$, $\overline{GM}$, angle of list and range of stability for weak side.

8-6. Rework example 8-3 for a draft of 50 feet and a $\overline{KG}$ of 43 feet. For this draft, at what angle of heel is the assumption of wall-sidedness no longer valid?

8-7. Write a computer program to generate a $\overline{GZ}$ curve for wall-sided ships up to the point of deck-edge immersion.

Submarine Hydrostatics, Stability, and Trim

9-1 Definitions

In order to maintain stable equilibrium (neutral buoyancy) between the force of gravity (weight) acting downward on a submerged submarine and the net hydrostatic force acting upward (submerged displacement), the crew of a submarine must actively maintain a state of static equilibrium. A surface ship automatically maintains a state of static equilibrium through a change in draft (displacement) to counteract weight additions and removals. The submerged displacement of a submarine, however, is fixed by the geometry of the hull and the density of the sea and does not depend on the actual weight of the submarine and its contents.

STATIC WEIGHT AND BUOYANCY CONTROL

The *submerged displacement* may be found by making appendage corrections to the displacement of the “molded envelope” of the visible hull of a submarine (figure 9-1). *Negative appendages* are those recessed volumes within the molded envelope that flood freely and cannot be blown down. Examples of negative appendages include the free-flooding bow and stern areas and the regions shown in figure 9-2. The volume of these negative appendages is subtracted from the volume of the molded envelope when computing the submerged displacement. *Positive appendages* are defined as the non-free-flooding portions of the submarine external to the envelope plus the volume of the hatches, hatch trunks, shafting, torpedo tubes, etc., inside the free-flooding areas. Examples of positive appendages external to the molded envelope include the rudders, stern planes, propeller, fairwater planes, and any non-free-flooding portions of the sail (fairwater). The volume of the positive appendages is added to the volume of the molded envelope when computing the submerged displacement.

The *main water ballast*, part of the submerged displacement, is contained in main ballast tanks (MBTs) within the molded hull but is generally external to the pressure hull (figure 5-36). Its purpose is to enable the submarine to surface when the tanks are blown down. The size of the tanks determines the amount of freeboard (reserve buoyancy), while the longitudinal location of the tanks influences the submarine trim in the surfaced condition. The capacity of the main ballast tanks is determined by subtracting the various volumes located within the MBTs, such as negative appendages (figure 9-2), buoyant volumes like air flasks, residual water that cannot be blown, and lead ballast (figure 9-3), from the gross volume of the main ballast tanks.

Light ship is the total weight of all the permanent components of the submarine. It is fixed at the time of construction and changes only when alterations are made to the ship.

Lead ballast is used to provide a margin in the weight estimates for submarine construction, so that the ship's submerged weight may be made equal to its submerged displacement. To keep the weight low for transverse-stability reasons, some of the lead may be located in the main ballast tanks, where it decreases both the MBT capacity and the amount of residual water (figure 9-3).

Load to submerge is the difference between the light ship plus lead and the submerged displacement. The load to submerge consists of the diving ballast, the variable load, the variable ballast, and the residual water.

Diving ballast equals the net capacity of the main ballast tanks above the residual water plus the net capacity of any fuel ballast tanks used for diving ballast on diesel submarines. It is also equal to the reserve buoyancy in the surfaced condition.

Variable load consists of the crew, cargo, and consumables such as fuel, provisions, fresh water, weapons, etc., that are not a permanent part of the submarine.

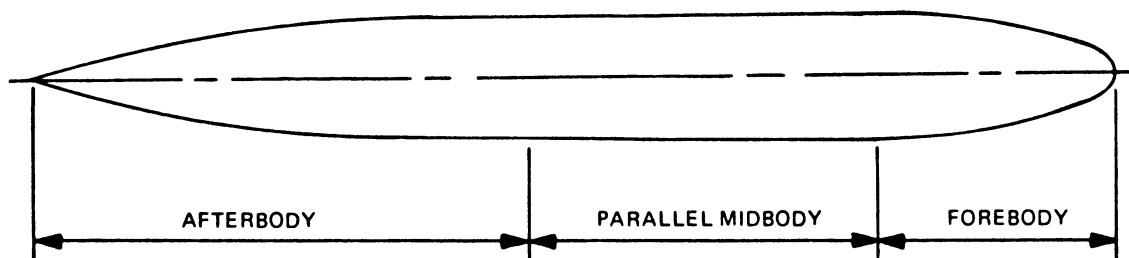


Figure 9-1. Molded envelope. (Courtesy of Electric Boat.)

Variable ballast is included in the load to submerge in order to account for changes in the variable load and changes in the density of seawater. By separating the variable ballast tanks along the length of the submarine, seawater may be admitted or discharged to correct for deviations from the neutral buoyancy condition and may be shifted fore or aft to correct for out-of-trim conditions (figure 9-4).

DYNAMIC DEPTH AND TRIM CONTROL

In addition to the variable ballast tanks for static depth and trim control at low speeds, wing-like control sur-

faces are used on submarines to provide more responsive control of depth and trim. They are effective at speeds greater than about 3 knots.

Fairwater planes are lifting surfaces located on the fairwater (sail) of a submarine whose primary purpose is depth control and whose secondary purpose is trim control.

Stern planes are horizontal ship control surfaces, located just forward or aft of the propeller, whose primary purpose is attitude (trim) control and whose secondary purpose is depth control.

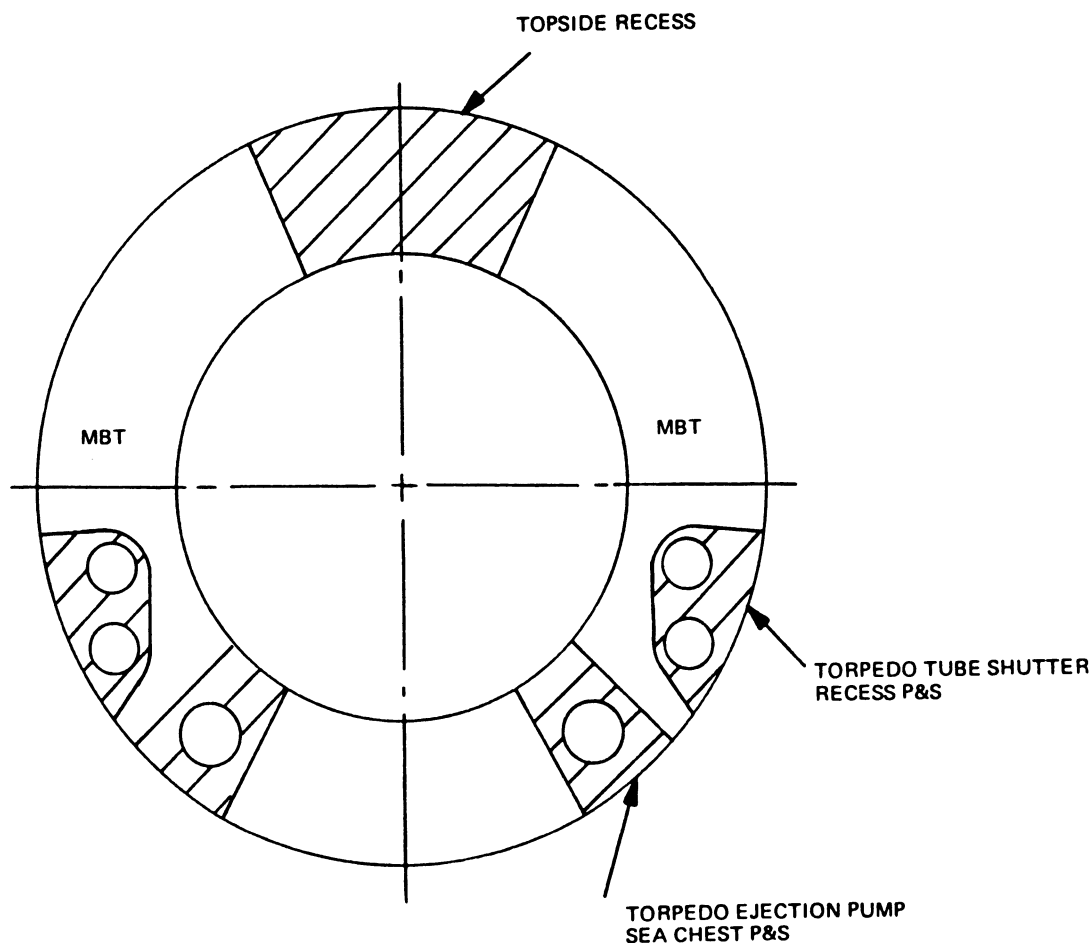


Figure 9-2. Typical negative appendages. (Courtesy of Electric Boat.)

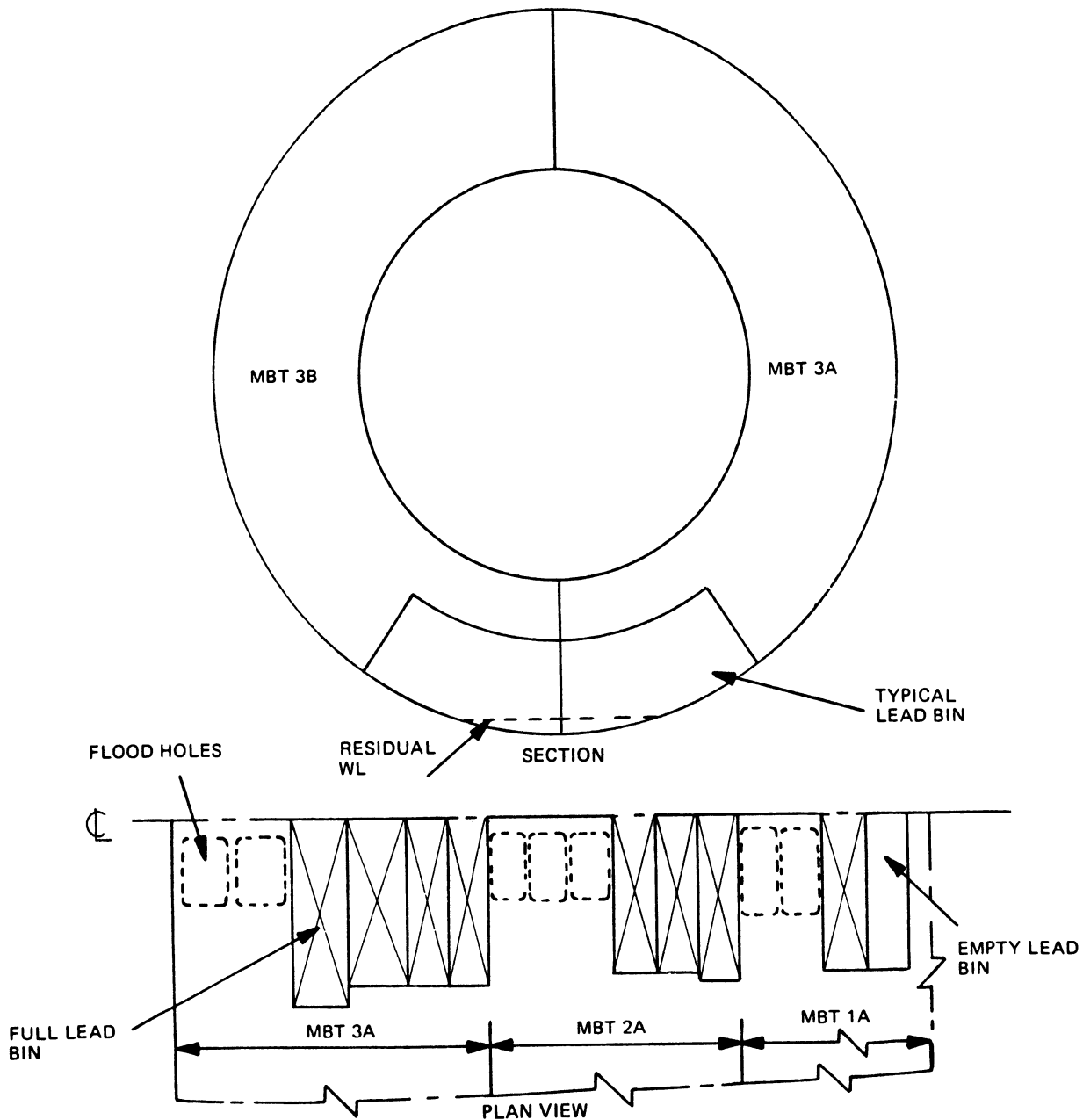


Figure 9-3. Lead ballast and residual water. (Courtesy of Electric Boat.)

9-2 Hydrostatics and Stability of Submersibles

A submarine operating on the surface exhibits the characteristics of a surface ship with some notable differences. Fleet-type submarines of World War II were really surface ships that could submerge. They had reserve buoyancies in excess of 30 percent of their submerged displacement and considerable waterplane area for surface stability. Modern nuclear submarines are "true" submersibles that must occasionally operate on the surface. These submarines have reserve buoyancies of only about 11 percent of submerged displacement and a very reduced waterplane area (see figure 1-10). Since both types of submarines depend on the center of gravity being below the center of buoyancy for sub-

merged transverse stability (figure 9-5), their range of stability extends to nearly 180° and the maximum righting arm occurs near 90° . A typical cross curve of stability for a nuclear submarine is illustrated in figure 9-6. Figure 9-7 shows a comparison of the righting moments for a World War II-type submarine, both submerged and on the surface, with those of a destroyer of approximately the same displacement.

It will be recalled that stability characteristics of all surface ships depend on two things:

1. *The underwater form and the positions of B and M.* Surface ships of normal form have comparatively high freeboard and straight sides with some flare, particularly near the bow. For a favorable position of G,

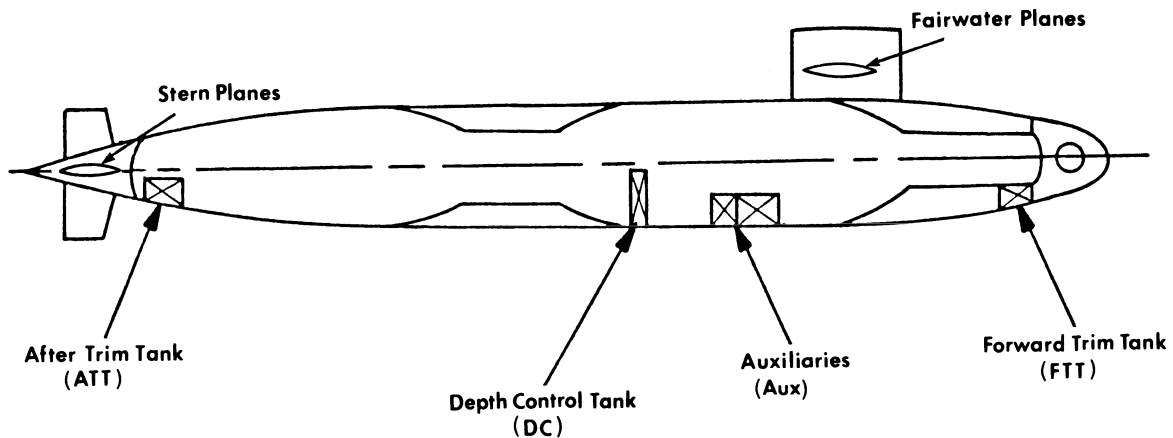


Figure 9-4. Depth- and trim-control tanks and surfaces. (Courtesy of Electric Boat.)

large righting arms and a good range of stability are obtained. This is because the waterplane area will increase through a fairly wide range of angles of inclination (until the deck edge is immersed), causing B to shift outwards to a greater degree than if the waterplane area was relatively constant. A correspondingly greater development of righting arm is realized.

Conversely, the nuclear submarine has a low freeboard and is nearly circular in section. There is no appreciable increase in the waterplane area as the angle of inclination is increased.

2. *The vertical distribution of weights.* As was observed previously, heavy weights low in a submarine result in the center of gravity G being located lower

than it is in a comparable surface vessel. A low G is the principal factor in overcoming the adverse effects of the submarine's form on stability. To insure a low G , lead ballast is usually placed in the bottom of the MBTs, its weight being about 3 percent of the submerged displacement. For a given B and M , the effect of lowering G is to improve the overall stability by realizing a very good range of stability, although the righting arms are still relatively small. In figure 9-7, note that the submarine's poor hull form on the surface and low center of gravity combine to produce a low, flat curve having a much greater range than the surface ship.

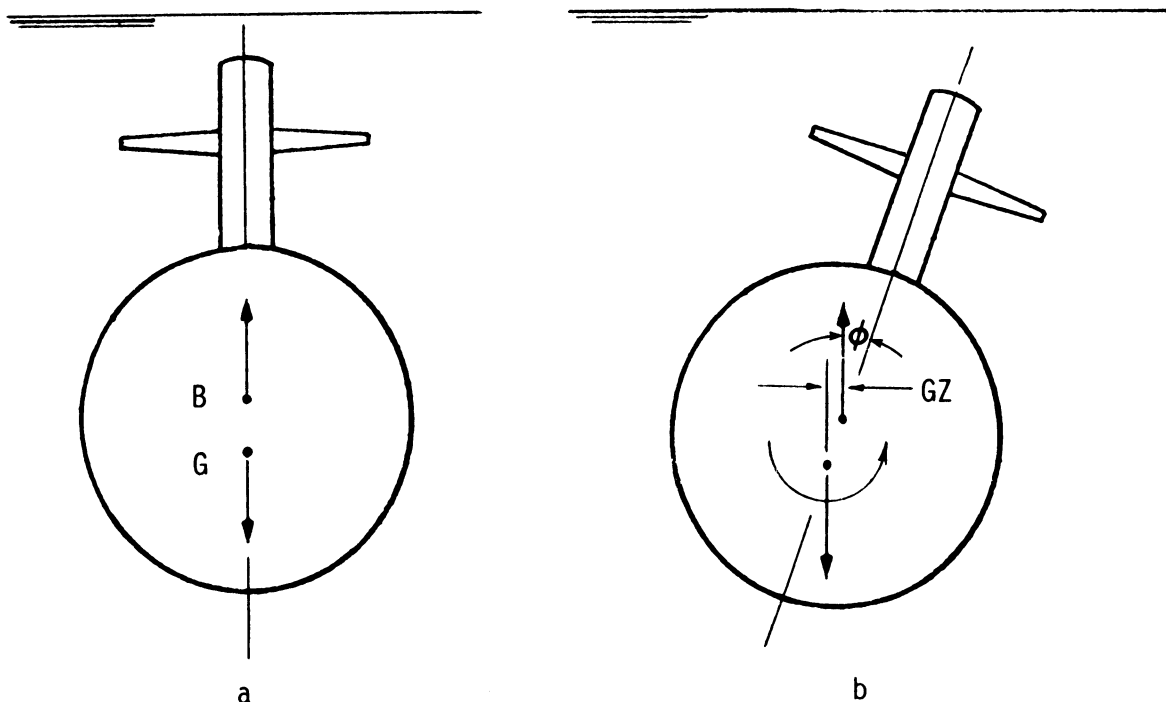


Figure 9-5. Righting arm for submerged transverse stability

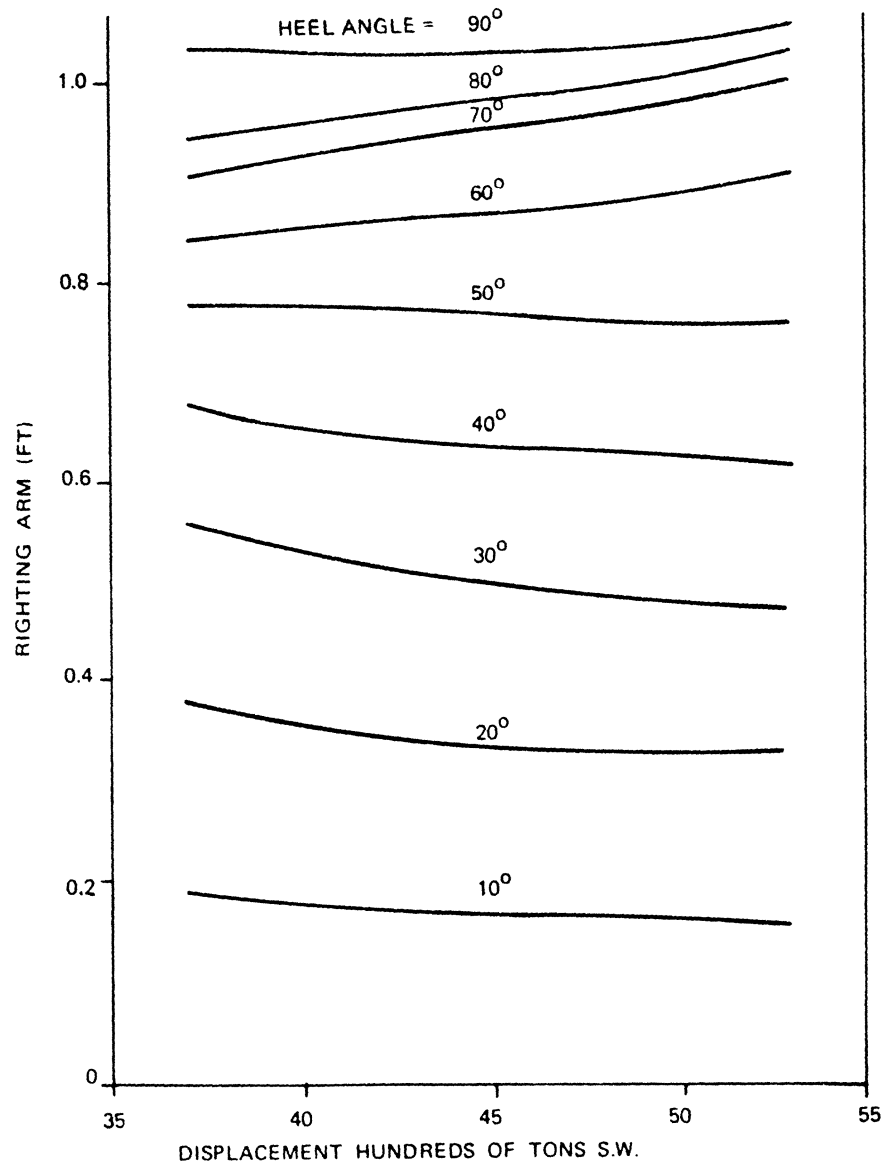


Figure 9-6. Cross curves of stability for a submarine. (Courtesy of Electric Boat.)

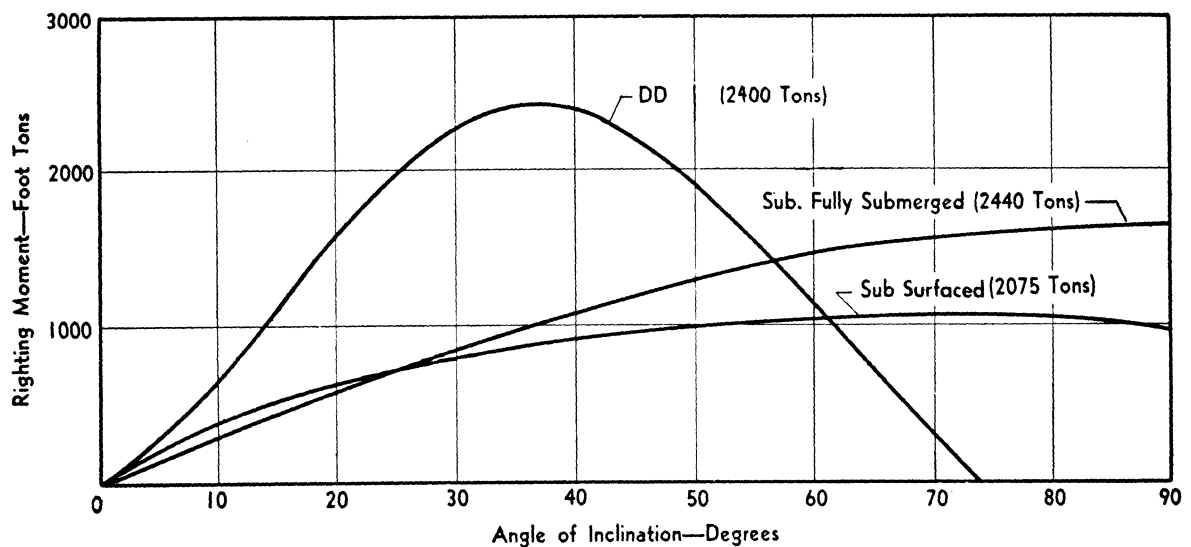


Figure 9-7. Curves of static stability for DD and typical submarine

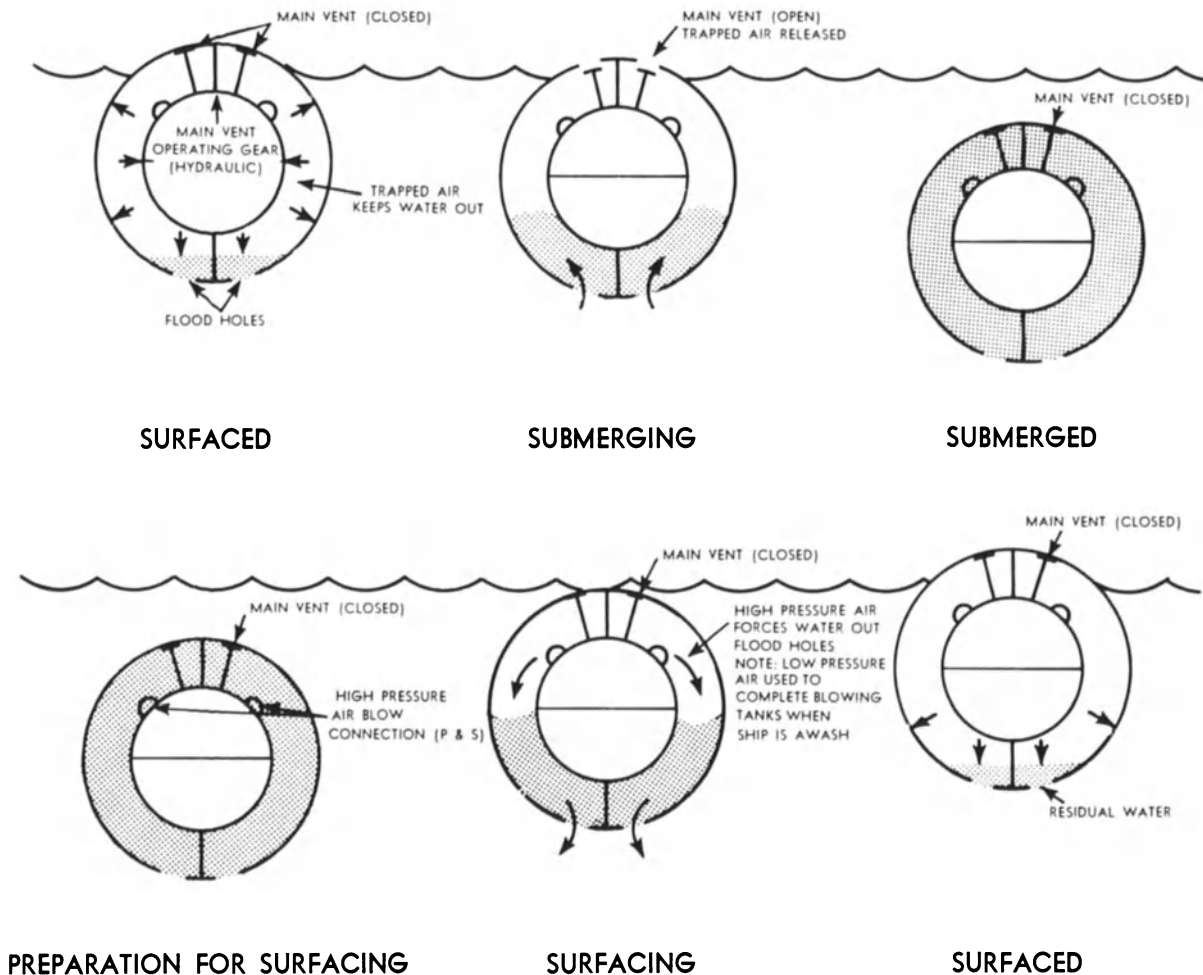


Figure 9-8. Use of the main ballast in diving and surfacing

9-3 Diving, Surfacing, and Their Effect on Transverse Stability

Diving a surfaced submarine is accomplished by opening the main vents to the main ballast tanks, as illustrated in figure 9-8. This lets trapped air out of the top of the MBTs and seawater in through the flood holes in the bottom of the tanks. As the seawater floods in, the weight of the submarine increases, but the net buoyant force (displacement) also increases as the freeboard is reduced. Nuclear submarines stay in a state of nearly neutral buoyancy throughout a low-speed dive and thus submerge very slowly. (Since submarines had to submerge often and sometimes very quickly in World War II, they had a *negative tank* specifically designed to give them negative buoyancy during the dive. As soon as the ship was fully submerged, water was blown out of this tank.) Once submerged, the main vents are closed, and the variable ballast tanks are used to establish a condition of stable equilibrium (neutral buoyancy).

Surfacing a submerged submarine is accomplished by admitting high-pressure air to the main ballast tanks (figure 9-8). This forces the water out through the flood holes in the bottom, decreasing the weight of the sub-

marine. It becomes positively buoyant and rises to the surface. Once the ship breaks the surface of the water, displacement decreases until the force of buoyancy equals the weight of the submarine and the ship assumes its surfaced draft (displacement).

TRANSITION STABILITY

During diving and surfacing operations, B , M , and G are not fixed but vary systematically. It is extremely important to know how the locations of these points vary with respect to each other in order that the minimum metacentric height during this transition period can be determined. To facilitate this study, an adequate number of intermediate conditions between surface, diving trim, and fully submerged states is calculated to permit the plotting of the curves shown in figure 9-9 for a World War II submarine. (These curves are *not* representative of a nuclear submarine.)

1. *Surface, diving trim with maximum fuel* (condition M). The submarine in this condition behaves like a surface ship as far as stability is concerned. Elaborate weight studies are made to locate G . Knowing the dis-

placement, B and M can be found from the curves of form.

2. *Trimming down.* As a result of the vessel's sinking to a higher waterline, a chain of events occurs. The waterplane area decreases; $\overline{BM}$ decreases; B moves upward, but at a slower rate than that at which $\overline{BM}$ is decreasing; therefore, M moves downward. G moves downward as the result of flooding the lower portion of the MBTs. However, the large amount of free surface present in the partially flooded MBTs and elsewhere causes a virtual rise of G to $G_{(v)}$. The locus of $G_{(v)}$, plotted as the dotted line, is the effect curve in making this stability study.

3. *Trimming down.* The tank tops are in the process of being submerged, and the reduction of waterplane area is proceeding at an increasing rate as the process of trimming down proceeds. Consequently, $\overline{BM}$ is being reduced at a greater rate, this being reflected by the increasing negative slope of the $\overline{KM}$ curve. B continues to rise and G (actual) is now below B . In this condition, it is assumed that the position of G and the amount of free surface are such that $G_{(v)}$ is also below B . The minimum value of $\overline{GM}$ occurs shortly after condition three (figure 9-9).

U.S. Navy submarines are designed to have a positive minimum $\overline{GM}$.

The value of $\overline{GM}$ is always minimum immediately

upon surfacing. If the submarine surfaces with a strong beam wind or sea, she will often assume a list. List control valves are placed in the low-pressure blow lines so that the boat can be righted by reducing the blowing of main ballast tanks on the side opposite the list. As water is blown from the MBTs' reducing free surfaces, $\overline{GM}$ increases and stability improves.

4. *Fully submerged.* This condition forms the end points for the curves.

It is assumed that the MBTs are completely flooded. Note that the $\overline{KG}_{(v)}$ and $\overline{KG}$ curves are not coincident in this condition. The free surface in the MBTs is now zero, but free surface present in the vessel will continue to be effective in causing a departure of the two curves.

With reference to the $\overline{KM}$ and $\overline{KB}$ curves in the vicinity of this condition, it should be noted that they are actually separate curves. Although this separation is too minute to indicate, it will exist as long as there is any structure of the submarine above the surface to provide a waterplane. Actual coincidence of the curves can occur only when the vessel is fully submerged and $\overline{BM}$ is reduced to zero.

SUBMERGED STABILITY

As implied above, a fully submerged vessel has no waterplane area. Thus I , and consequently $\overline{BM}$, become zero. B and M are now coincident points, and $G_{(v)}$ must

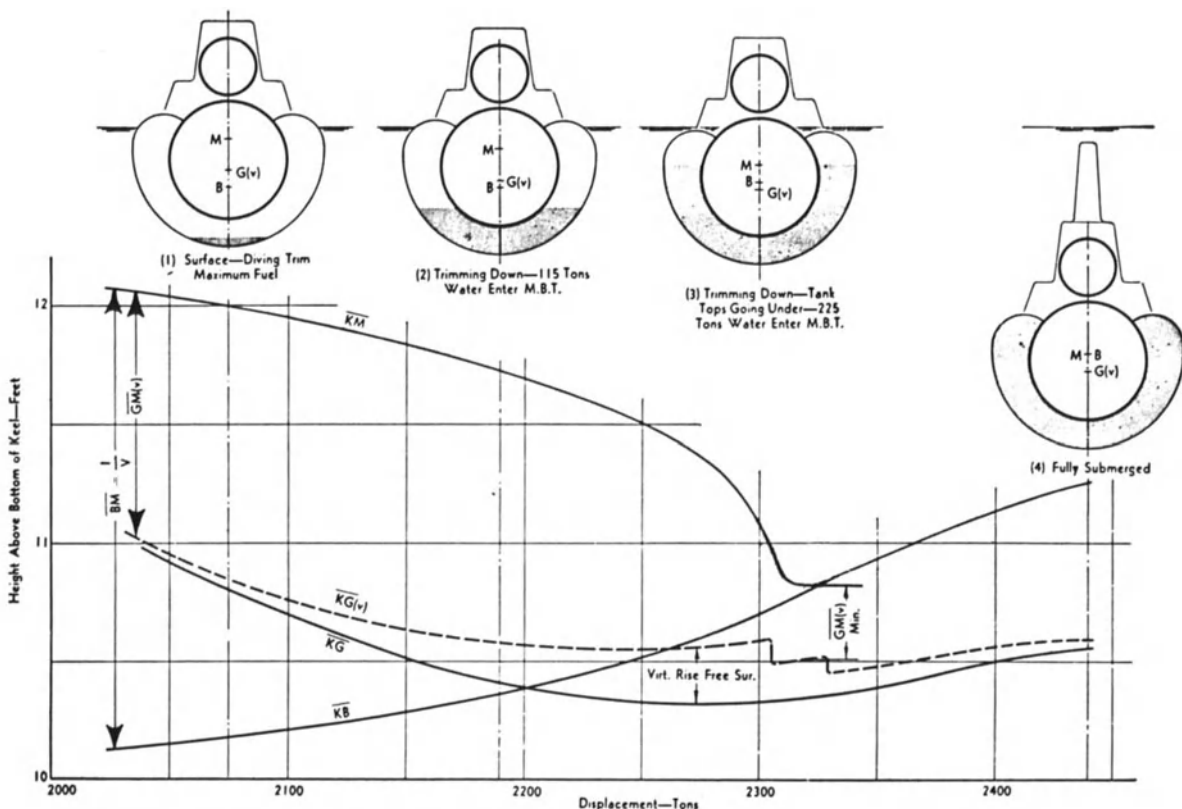


Figure 9-9. Trimming down curves for condition M

be below B . The magnitude of $\overline{BG}_{(v)}$ becomes the criterion of submerged stability, and the equation of the submerged static-stability curve, from figure 9-5, is

$$\text{righting arm } (\overline{GZ}) = \overline{BG}_{(v)} \sin \phi \quad (9-1)$$

The equation for righting moment about G is

$$\text{righting moment} = \Delta_s \overline{BG}_{(v)} \sin \phi \quad (9-2)$$

A comparison of righting moments of the submarine on the surface and submerged is made in figure 9-7.

9-4 Longitudinal Stability and Trim Control

LONGITUDINAL STABILITY

As in the case of surface ships, the longitudinal stability of a surfaced submarine is of little concern owing to the very large values of the longitudinal metacentric height $\overline{GM}_L$. However, as the waterplane vanishes when the vessel submerges, $\overline{BM}_L$ is reduced to zero and M_L becomes coincident with B and M . In other words, the actual transverse and longitudinal metacentric heights are equal.

$$\overline{GM} = \overline{GM}_L = \overline{BG}$$

But recall that stability considerations are based on the virtual height of the center of gravity $G_{(v)}$. $\overline{G}_{(vT)}M$ is greater than $\overline{G}_{(vL)}M$, where the subscripts vT and vL denote transverse and longitudinal locations of the virtual center of gravity. As in transverse stability, the magnitude of $\overline{BG}_{(vL)}$ becomes the criterion of longitudinal stability. Hence,

$$\overline{BG}_{(vT)} > \overline{BG}_{(vL)}$$

As may be suspected, the reason for this inequality is that the effect of free surface in decreasing transverse stability is less than the effect of free surface in reducing longitudinal stability. The longitudinal dimension of most tanks and bilges in a submarine is greater than the transverse dimension. Determination of longitudinal free-surface correction requires calculating the moments of inertia of tanks and bilges about their athwartship axis and cubing the larger of their two dimensions. It is possible for a free surface having only a small effect on transverse stability to make the vessel unstable longitudinally. In the interest of submerged control of the vessel, it should be evident that free surface must be reduced to a minimum. Particular attention should be given to the MBTs. It is possible for air bubbles to form at the top of the tanks due to leaky blow valves, causing the introduction of free surface as well as a loss of main ballast weight, which results in the ship becoming light. The MBT vents can be opened periodically (cycled) to allow any air bubbles to escape. An effort should also be made to keep the bilges reasonably dry.

STATIC TRIM CONTROL FOR SUBMARINES

For simplicity, a profile of a submarine showing two general groups of ballast tanks, auxiliaries, and forward and after trim tank (located inside the pressure hull) will be used to illustrate static trim control. It is assumed that the submarine has no way on and that control of the submerged submarine can be maintained by shifting, adding, or removing weight. The most convenient form of weight to use is seawater, because of the facility with which it can be moved by means of pumps or air pressure and directed through a trimming manifold to the desired location.

dotted line vector = individual weight and buoyant forces

solid line vector = force resultant

W = weight of submarine

W_s = weight submerged at neutral buoyancy

W_1, W_2, W_3 = resultant weight for specific condition

w = weight added or removed

w (with subscript) = weight of seawater in a particular tank

Δ = buoyant force on surface in diving trim

Δ_s = buoyant force submerged

b = buoyant force of reserve buoyancy

Since submerged trim is far more sensitive to longitudinal weight shifts than surface trim, it is important to operate on the surface in diving trim, so that the submarine may be readily submerged in a condition of static equilibrium. Some submarines operate on the surface in level trim, but for others it is desirable to trim slightly by the stern in order to keep the propeller adequately submerged, as illustrated in figure 9-10(a). For this type of operation the combined center of gravity of the diving ballast in the MBTs (w_{MBT}) is located forward of the submerged center of buoyancy B_s , as shown in figure 9-10(b). As the tanks are blown for surfacing, the ship's LCG is driven aft until the surfaced LCG is on the same vertical axis as the surfaced LCB. This requires that the submarine be designed with the center of buoyancy b of the vessel's emerged volume when surfaced (that is, the reserve buoyancy) located forward of the submerged center of buoyancy, but slightly aft of the combined center of gravity of the diving ballast, as illustrated in figure 9-10(b). For those submarines that operate on the surface in level trim, the center of buoyancy of the emerged volume is directly above the combined center of gravity of the MBTs. This ensures that the LCB and LCG will shift equal longitudinal distances in going from surface to submerged operation and vice versa.

Up to this point, only a submarine at the desired weight and trim has been discussed. A submarine is considered to be out of trim when the force of gravity is not equal and opposite to the force of buoyancy and/

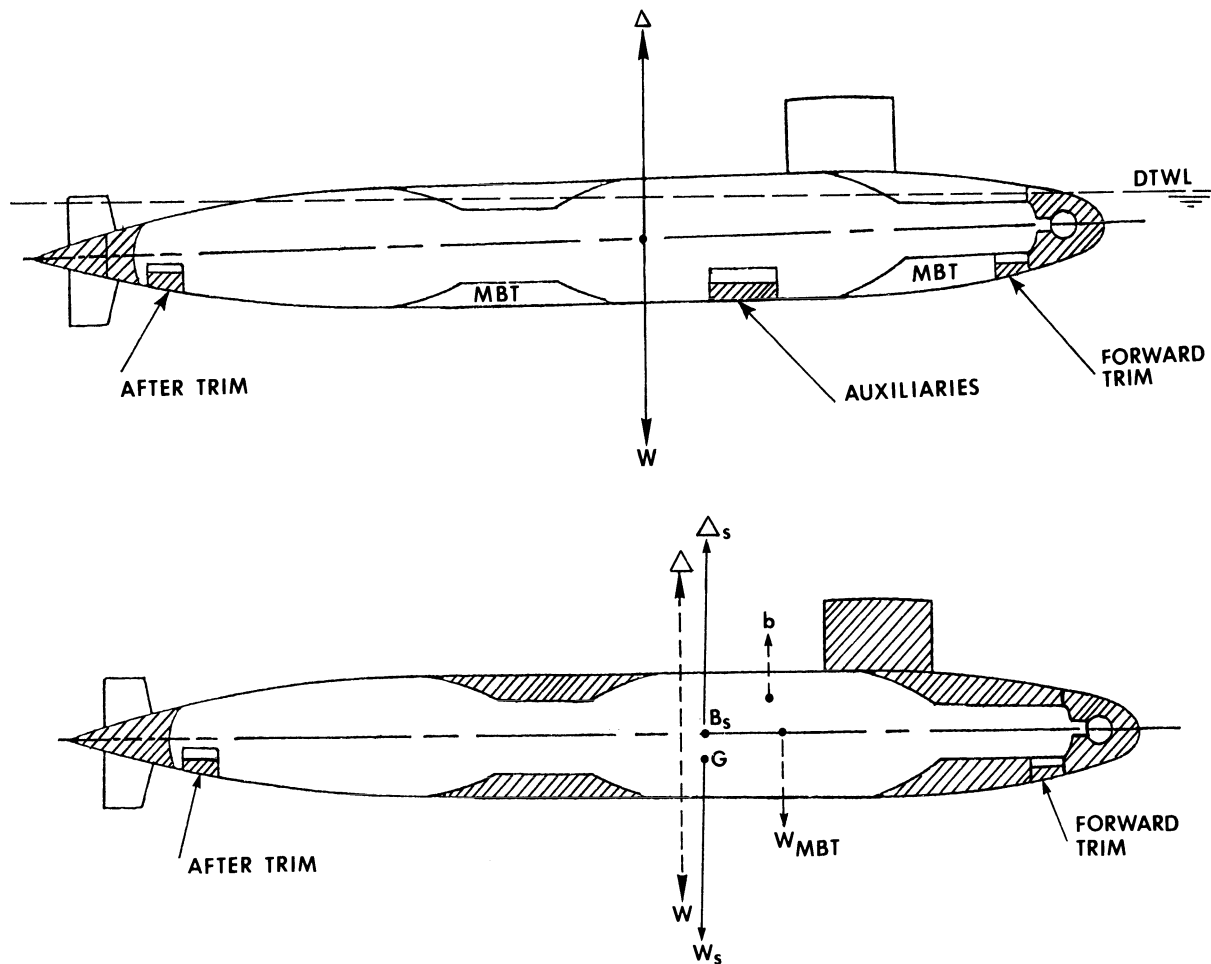


Figure 9-10. Submarine designed for trim by the stern when operating on the surface in diving trim (distances exaggerated for clarity): (a) surfaced in diving trim, down by the stern (b) submerged, neutral buoyancy and zero angle

or acting in the same vertical plane. Trimming a submarine is accomplished by shifting seawater (variable ballast) within the ship or taking on or discharging seawater ballast from the appropriate variable ballast tanks. Trim analysis is made easier when the auxiliary tanks are located on a vertical axis with the LCB, but this is not the case for many modern submarines. The forward trim tank is frequently smaller than the after trim tank because of space limitations; this requires that the auxiliary tanks be located forward of the LCB (figure 9-10). These tanks must be used to compensate for weight changes and shifts in the center of gravity of the submerged submarine caused by change of location of crew, stores, weapons, the consumption of fuel oil and potable water, the firing of weapons, changes in the seawater density, etc. There are eight possible out-of-trim conditions caused by combinations of the above changes. These conditions are classified by the U.S. Navy as

1. Heavy Overall/Okay Fore and Aft (HOA/OK F&A)
2. Heavy Overall/Heavy Forward (HOA/HF)
3. Heavy Overall/Heavy Aft (HOA/HA)

4. Light Overall/Okay Fore and Aft (LOA/OK F&A)
5. Light Overall/Light Forward (LOA/LF)
6. Light Overall/Light Aft (LOA/LA)
7. Okay Overall/Light Forward/Heavy Aft (OKOA/LF/HA)
8. Okay Overall/Light Aft/Heavy Forward (OKOA/LA/HF)

Notice that both the overall conditions and the fore and aft conditions have been described. No inconsistencies are allowed; that is, a ship cannot be heavy overall and light forward. If the ship is heavy overall it must be heavy somewhere (fore/aft or at the center of gravity).

The submarine is considered to be "gross out of trim" when it must be angled to the horizontal using the stern planes at one-third speed in order to maintain depth. This is generally the result of one of the first six out-of-trim conditions and indicates that the forces of gravity and buoyancy are not balanced. To correct the trim condition,

- | <i>If the ship is</i> | <i>then</i> |
|-----------------------|------------------|
| 1. HOA/OK F&A | pump aux. to sea |
| 2. HOA/HF | pump FTT to sea |

3. HOA/HA pump ATT to sea
4. LOA/OK F&A flood aux. from sea
5. LOA/LF flood FTT from sea
6. LOA/LA flood ATT from sea

These corrections are only approximate if the center of gravity of the weight causing the heavy or light condition overall is not located directly at the center of gravity of the ballast tank. For example, correcting the HOA/HA condition shown in figure 9-11 by pumping w tons from the ATT to sea will result in condition eight above (OKOA/LA/HF), which must be further compensated by transferring ballast aft from a forward tank.

Trim conditions seven and eight are considered to be "close in trim" and may be corrected by transferring seawater ballast from one variable ballast tank to another. Since the metacenter is now coincident with the center of buoyancy B , equation 7-9 becomes

$$wl = \Delta \overline{BG} \tan \theta \quad (9-3)$$

This equation can be used to solve for the weight of seawater that must be shifted to compensate for a trim angle θ when $\overline{KB}$ and $\overline{KG}$ are known.

Example 9-1: A submarine with a submerged displacement of 4,800 tons has a $\overline{KB}$ of 15.9 feet and a $\overline{KG}$ of 14 feet. How much seawater must be transferred from the after tank to the forward trim tank to correct for a trim angle of 2° if the distance between the tanks is 222 feet?

Solution:

$$\begin{aligned}
 \overline{BG} &= \overline{KB} - \overline{KG} \\
 &= 15.9 - 14 \\
 &= 1.9 \text{ ft} \\
 w &= \frac{\Delta \overline{BG} \tan \theta}{l} \\
 &= \frac{4,800(1.9) \tan 2^\circ}{222} \\
 &= 1.43 \text{ tons} \\
 &= \boxed{3,210 \text{ pounds}}
 \end{aligned}$$

9-5 Factors Affecting Practical Submarine Statics

Before leaving the subject of submarine statics, it would be well to call attention to the following facts that complicate the operations of a submerged submarine. Consider the equation

$$\begin{aligned}
 \text{weight} &= \text{buoyant force} \\
 \text{submerged} &= \text{submerged} \\
 \text{or } W_s &= \Delta_s \quad (9-4)
 \end{aligned}$$

This equation must be satisfied if the submerged submarine is to remain in neutral buoyancy. Also consider the equation

$$\begin{aligned}
 \text{buoyant} &= \text{volume of water} \\
 \text{force} &= \text{displaced times the} \\
 \text{submerged} &= \text{specific weight of water}
 \end{aligned}$$

$$\text{or } \Delta_s = pg \Delta_s$$

The factors on the right side of the equation are *not* constant and are best studied separately.

Volume of Water Displaced. As a submarine submerges, the hydrostatic pressure increases, and at greater operating depth, an appreciable amount of the volume of displacement is lost due to the compression of the hull. Thus, holding the density constant, Δ_s is decreased as the ship goes deeper, and a heavy-overall condition will exist. Since the operating personnel have control over weight primarily, W can be decreased by pumping out variable ballast water until W_s again equals Δ_s . The volume of water displaced (and hence Δ_s) can be altered to a very small degree by raising or lowering the periscopes and various masts. Often the minute change in Δ_s will be sufficient to keep the submarine in a state of neutral buoyancy without changing the weight of the ship.

Density of Water. The density of seawater depends primarily on two factors: temperature and salinity. As the temperature decreases, the density increases. Conversely, as the salinity decreases, the density decreases. In the open ocean the salinity varies but little and can be considered constant. The temperature, however, is a variable, usually decreasing at different rates as the depth increases, depending on the location. Submarines carry instruments that trace the temperature gradient with changes in depth; such records are of invaluable assistance in planning future operations. Because of the temperature decrease, the density of the seawater will increase as the submarine goes to greater depths. In the above equation, if the volume of water displaced is held constant, the effect of increasing depth would be to increase the buoyant force. Therefore, Δ_s would exceed W and the vessel would become light overall. Weight must be added in the form of variable ballast water to make W_s equal Δ_s . If both factors of the right side of the above equation are variables, they will tend to counteract each other. Although the vessel's hull is compressed as it goes deeper (decrease in Δ_s), if the smaller amount of water displaced weighs more than the larger amount of water displaced near the surface, the combined effect would cause the submarine to become light, and variable ballast water would have to be taken aboard. In the vicinity of mouths of fresh water rivers and in

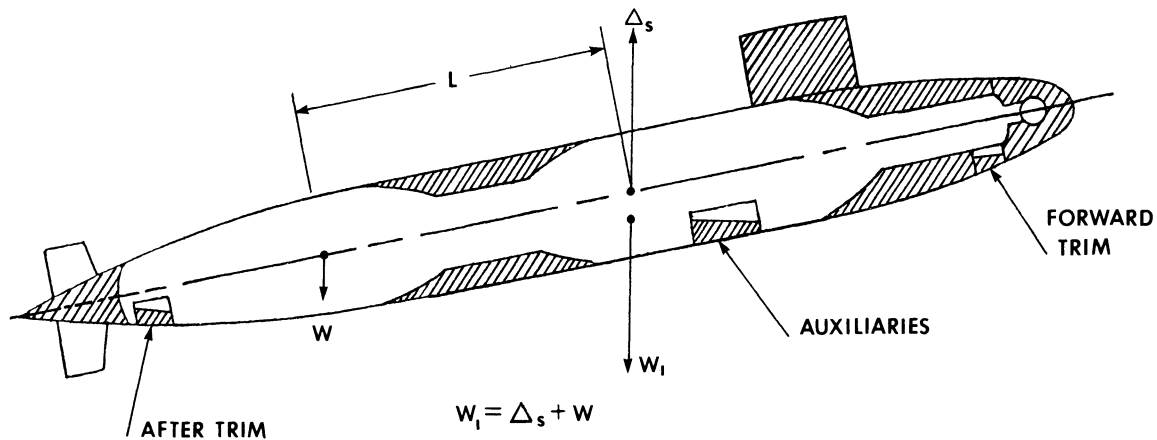


Figure 9-11. Heavy overall/heavy aft condition caused by a weight added aft

various other parts of the ocean, the salinity is far from constant. This presents a second variable in density determination and further complicates correct ballasting.

These facts in no way invalidate the basic concepts developed in this section. To be in a state of neutral buoyancy, the total weight of the submarine must equal the weight of the water it displaces.

The fact that the ocean is, in effect, divided into layers of water of varying densities, owing mainly to temperature differences, aids the submariner but com-

plicates the life of diving officer. Ships can actually rest on layers with very little or no headway on, an advantage when rigged for silent running. Layers also offer protection from detection by surface craft because they deflect the stream of acoustic impulses from sonar gear.

9-6 The Equilibrium Polygon

As indicated in the foregoing, it is necessary on submarines operating below the surface to compensate for variations in longitudinal moments and weight by mak-

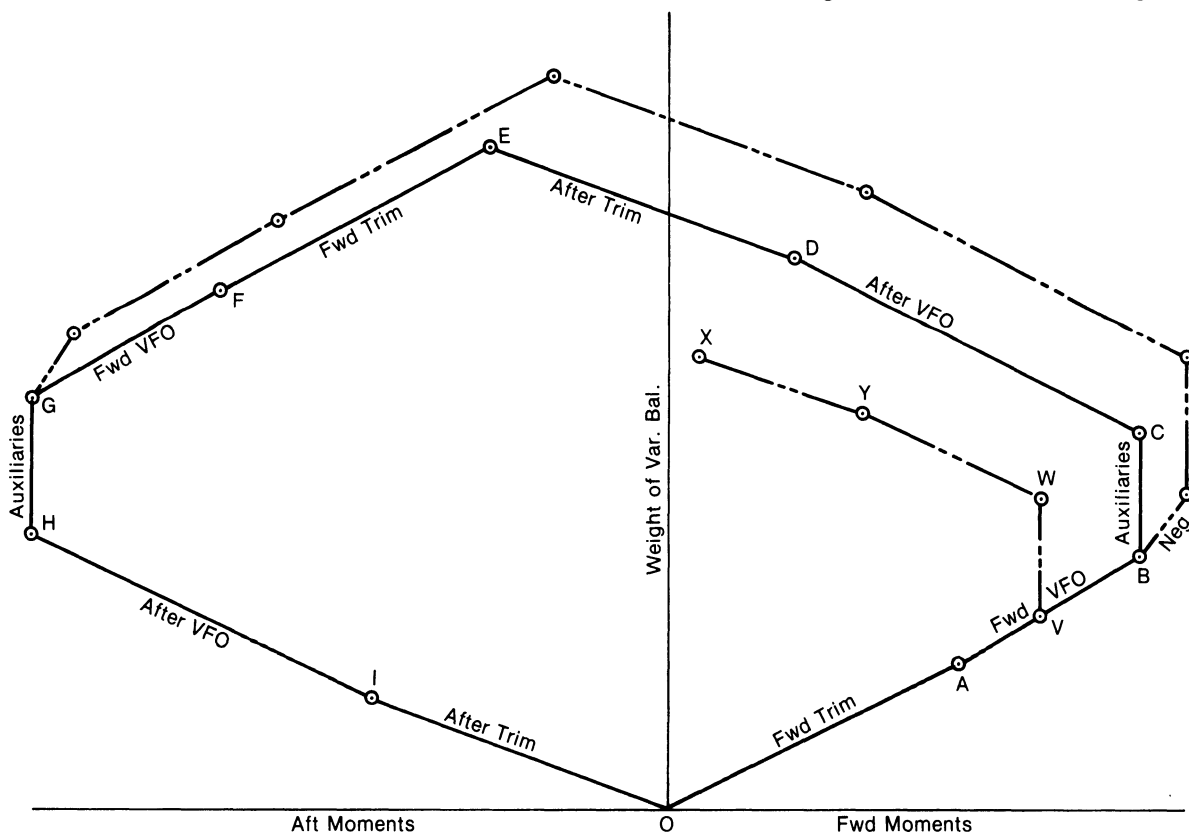


Figure 9-12. Equilibrium polygon for submarine with auxiliary tank located at LCB

NOTE: SEA WATER DENSITIES INDICATED IN POUNDS PER CUBIC FOOT. TONS ARE LONG TONS SALT WATER.

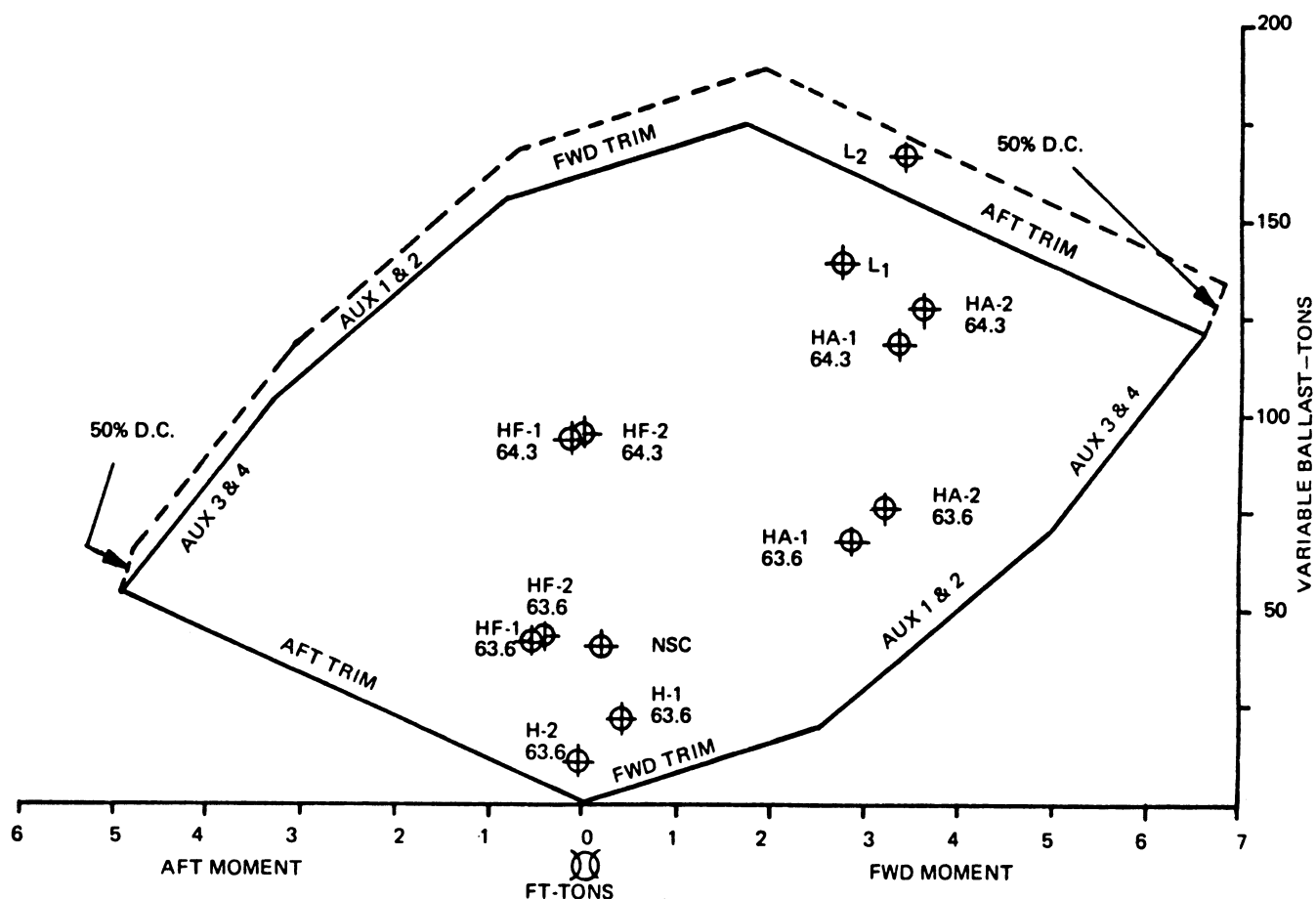


Figure 9-13. Equilibrium polygon for a submarine with auxiliary tanks forward of LCB. (Courtesy of Electric Boat.)

ing adjustments in the variable ballast. This ongoing daily problem can be somewhat simplified or mechanized by a graphical plotting method called the equilibrium polygon.

Referring to figure 9-12, which is a typical example of this graphical method of accounting, note that the horizontal dimension represents variable ballast moment about the trim axis of the submerged submarine. (The axis location is here designated O .) The vertical dimension or scale represents the weight of variable ballast. Thus, each side of the polygon represents the effect of filling one of the variable ballast tanks. These tanks are identified along the appropriate side of the polygon, here shown for a typical diesel submarine with the auxiliary tank located at the LCB.

For example, the line OA shows the development of the weight of ballast together with its moment as the forward trim tank is filled. Proceeding and adding to this weight and moment summation as the forward variable fuel oil tank is filled, point B is reached. Next, filling the auxiliaries, which are located very near the

submerged center of buoyancy (trimming axis), the moment change is negligible and the direction of BC is therefore vertical. The after variable fuel oil tank, when filled, reverses the direction of the moment, and the line of the developing polygon moves left from C to D . Finally, with the filling of after trim, point E is reached, which is the top of the polygon. It thus represents the total summation of added ballast weight and the net moment produced, either forward or aft of point O (in this case slightly aft). Point E could have been reached by plotting from the after trim tank to the forward instead of from forward aft as described above. If this direction is plotted as in figure 9-12, then the polygon is completed and closed. In this form it makes a useful tool. It should be noted that the tanks, when plotted from aft forward, will appear parallel to the polygon sides, showing these same tanks plotted from forward aft. Thus, side FE is parallel and equal in length to side OA , and so forth.

By using the equilibrium polygon, which is an operational tool as well as a design procedure, any desired

submerged condition may be reached if the calculated variable ballast required for a particular condition of loading falls within the polygon.

When using the equilibrium polygon the ballast weights of water taken on (added) are computed on the basis of the capacity of the tanks at the rate of 35 cubic feet per ton.

In figure 9-12, the broken line external to the polygon indicates the additional side used when the submarine is equipped with a negative tank; it is considered part of the variable ballast system. The negative tank as used produces a total submerged weight that exceeds the neutral submerged buoyancy, and thus the condition must be assumed as heavy overall and the submarine will sink.

Figure 9-13 shows an equilibrium polygon for the submarine illustrated in figure 9-4. This type of submarine has auxiliary and depth-control tanks located away from the LCB; thus, there are no vertical lines on the diagram. Since the depth-control tanks are closest to the LCB, they are very useful for correcting a heavy-overall or light-overall condition while minimizing the effect on the trim of the submarine.

Trim control using the fairwater and stern planes will be discussed in chapter 14.

Problems

9-1. You are a brand new diving officer on the USS *Tecumseh* (SSBN 628). Thirty men move from the crew's mess 30 feet forward of the ship's center of gravity to the forward torpedo room, 180 feet forward of the ship's center of gravity. How much water will you have to shift from the forward trim tank (175 feet forward of the center of gravity) to the after trim tank (200 feet after of the center of gravity) to compensate for this weight shift if the average weight of the men is 165 pounds?

9-2. Torpedoes are shifted on a submarine. Four MK 48 torpedoes weighing 4,000 pounds a piece are moved forward 25 feet and one MK 37 weighing 1,790 pounds is moved aft 30 feet. How many gallons of water must be shifted between forward and after trim tanks if the tanks are 270 feet apart?

9-3. A submarine is 300 feet long by 30 feet in diameter; it is submerged in salt water in neutral buoyancy with a center of gravity 13 feet above the keel. A weight of 20 long tons is shifted 20 feet athwartships. Calculate the angle of list.

9-4. Rework example 9-1 for a $\overline{KG}$ of 13.5 feet and a trim angle of 3° .

Ship Hazards and Vulnerability: Damaged Stability

10-1 Introductory Concepts

A ship exposed to the frequently hostile environment of the oceans is faced with many hazards, the most serious being the ultimate loss of reserve buoyancy, which results in sinking. A variety of events can lead to this catastrophe. The most serious contributive factors are loss of stability through uncontrolled flooding, resulting in capsizing and plunging, or structural failure, which may or may not cause the ship to capsize before sinking. The elements of structural adequacy were discussed in chapter 5; this chapter will be primarily devoted to hazardous and deteriorated buoyancy and stability situations.

There are, of course, other hazards that will be noted, some of which are not particularly related to the ultimate failures cited above. A ship may be reduced to a useless, uninhabitable hull by extensive fire or grounding. Because of an underwater explosion, a warship may also be rendered mechanically useless or, because of nuclear fallout and resulting radiation, may become uninhabitable and inoperable. These hazards and others must be considered by planners and designers.

WATERTIGHT SUBDIVISION

From the earliest days of shipbuilding, man has tried to prevent the flooding and sinking of his ship. Flooding became a matter of increasing concern with the larger ships constructed of iron and steel in the nineteenth and early twentieth centuries. As they continued to grow and as disasters occurred, such as the tragic sinking of the *Titanic* in 1912, international conferences of responsible government agencies and ship designers began to formulate rigorous international regulations and criteria for watertight internal subdivision to restrict flooding. These criteria for modern ships are presently met by specific flooding calculations and the concept of

floodable length. For introducing these concepts, the most important definitions are

1. *Bulkhead deck*—The bulkhead deck is the uppermost deck to which the transverse watertight bulkheads extend (usually the main deck).

2. *Margin line*—The margin line is a line drawn parallel to, and a minimum of 3 inches below, the bulkhead deck at the side. For ships with a stepped bulkhead deck, more than one margin line may be required.

3. *Permeability (volume)*—Permeability is the percentage of volume in a space that can be flooded and is expressed as

$$\mu = \frac{\text{available volume}}{\text{total volume}}$$

4. *Floodable length*—Floodable length is the maximum length at a given longitudinal position within a ship that can be symmetrically flooded at the prescribed permeability without sinking below the margin line.

5. *Factor of subdivision*—The factor of subdivision is an arbitrary factor applied to the floodable length to obtain the permissible length of compartments within a ship. The factor of subdivision is prescribed by national and international rules and conventions as a function of the length of the ship and the type of service (passenger versus cargo and type of cargo). Generally speaking, the factor of subdivision ensures that one, two, or three compartments must be flooded before the ship settles to the margin line. Ships designed to these rules are sometimes called one-, two-, or three-compartment ships with reference to their damaged-stability capabilities.

6. *Permissible length*—The permissible length of a compartment within a ship is obtained by multiplying the value of the floodable length at the center of the compartment by the factor of subdivision.

7. *Curve of floodable length*—The curve of floodable length is a curve that at every point in its length has an ordinate representing the length of ship that may be flooded with the center of length at that point and without the margin line being submerged.

The floodable-length curve is a valuable tool in assessing the ship's damaged-stability characteristics. A brief description of this curve and example will be helpful.

Assume that the floodable-length curve, shown in figure 10-1, has already been determined for a given permeability and drawn on the profile of the ship. Take a point, such as *A* on the curve, which may be at any distance *X* from the midship section $\overline{\overline{0}}$. The point *A*, then, is at a distance *l* above the base line of the plot, its ordinate distance, which corresponds to the length of the compartment whose center is at *X* and which, when flooded, would trim the ship to a new waterline WL_1 tangent to the margin line. Any further increase in the length of a compartment at this point would indicate a point of intersection above *A*, with flooding to a waterline above the margin line.

The process of calculating the values for plotting the floodable-length curve is a tedious one but can be greatly facilitated by use of a digital computer. The method and theory can be quickly summarized.

In order to determine the extent and position of flooding that will bring the ship from its operating waterline *WL* (figure 10-1) to the waterline WL_1 , tangent to the margin line, one uses an equation of moments about the original center of buoyancy, which yields

$$\Delta_1 \overline{BB_1} = w(x + \overline{\overline{0}B})$$

and

$$x = \frac{\Delta_1 \overline{BB_1}}{w} - \overline{\overline{0}B} \quad (10-1)$$

where Δ = original displacement

Δ_1 = final displacement

$w = \Delta_1 - \Delta$, weight of flooding water

B = original center of buoyancy

$\overline{\overline{0}B}$ = distance from *B* to amidships

B_1 = final center of buoyancy

x = distance from $\overline{\overline{0}}$ to center of flooded compartment.

The volume of salt water admitted is $35w$, and the total volume of the compartment is therefore

$$V = \frac{35w}{\mu} \quad (10-2)$$

Hand calculations for volumes of the ship's hull at various trimmed waterlines are made less tedious by using previously prepared curves called *Bonjean's curves*. These are a series of curves along the ship's profile that show, at each station, the sectional area of that section up to any waterline. It is possible, therefore, to plot any inclined or trimmed waterline on Bonjean's curves and then pick off the necessary sectional areas for computing the displacement at the waterline and *B*, through an iteration process. An example of a Bonjean curve is indicated in figure 10-2.

It must be pointed out that the floodable-length curve presented here represents only a basic concept. In actual design work and practice, full notice is taken of different permeabilities resulting from structural variations in individual designs, double-bottom voids, and

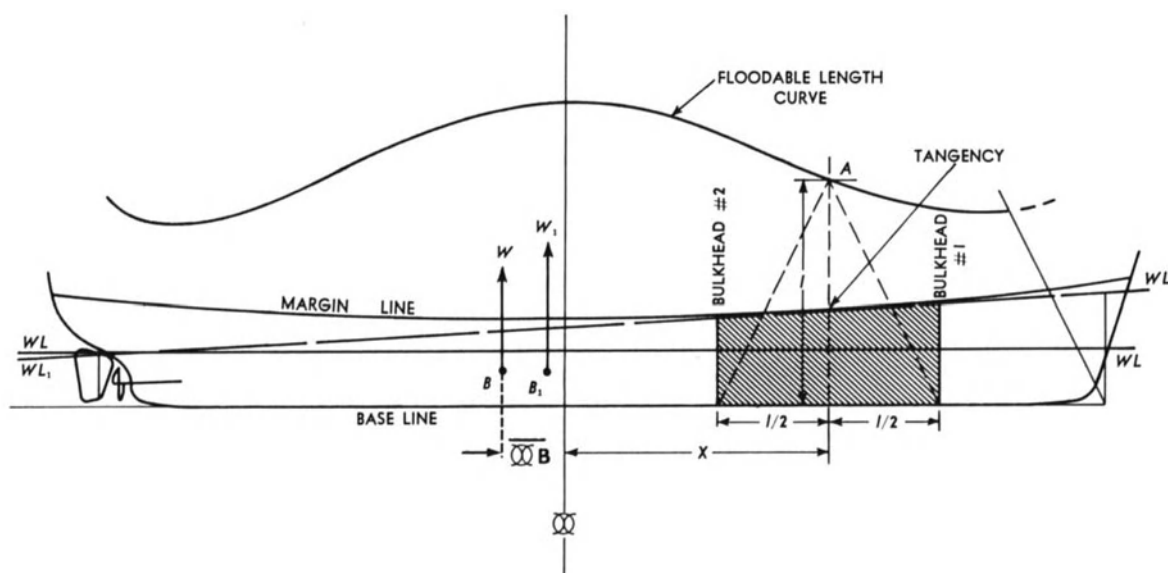


Figure 10-1. Curve of floodable length

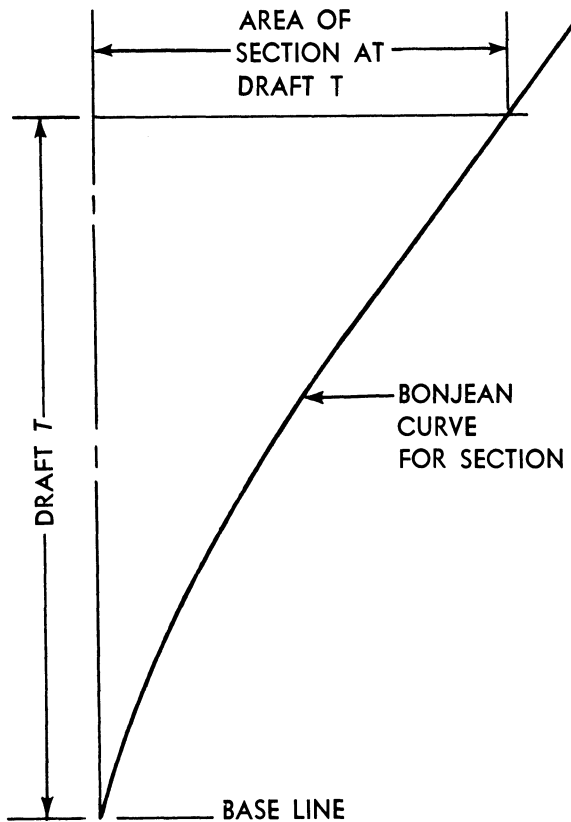


Figure 10-2. Bonjean curve

the contents of machinery spaces, cargo spaces, living spaces, etc.

In such flooding calculations, symmetrical flooding is taken into consideration, which of course allows for no transverse moment or resulting list. This factor is ultimately modified by superimposing upon the floodable-length curve another curve for floodable length for transverse stability that does deteriorate even under symmetrical conditions. A modified floodable-length curve is shown in figure 10-3.

Loose Water in Compartments

In preceding chapters, static-stability curves were introduced, and the effects of weight shifts and weight changes on the static-stability curves were discussed. It was implied that the center of gravity of the shifted or added weight remains fixed in the ship, regardless of her motion. In the case of solids, this is true if the material is properly secured to prevent shifting. In the case of liquids, this is true only if the liquid completely fills its container. If the compartment is only partially filled, the liquid moves from side to side as the ship rolls, because the surface of the liquid tends to remain horizontal. Water that partially fills a compartment, as a result of underwater damage, drainage, or fire fighting, is called *loose water*. The presence of loose water has a detrimental free-surface effect on both the initial and overall stability (as a free surface does in the regular tankage of a ship).

Surface Permeability

In any compartment in a ship, there are generally some solid objects that will project through the surface of flooding water or contained liquid. If these solid objects are made fast so that they will not float and are not permeable themselves, the free-surface effect of the liquid in the compartment will be diminished in proportion to the amount of free surface so reduced by the objects. In many ships' compartments, the amount of surface so reduced will depend upon the depth of the liquid in the compartment. In shipboard compartments, because of the many shapes and heights of the objects, the amount of surface suppressed will vary.

The *surface-permeability factor* is the ratio of the moment of inertia of an actual free surface to the moment of inertia of the same surface with no objects projecting through the surface. It is indicated by the symbol μ_s .

This factor is multiplied by the unrestricted free-surface moment of inertia i to obtain the actual free-surface moment of inertia. The factor used must be a matter of judgment, since its value changes with angle of heel, depth of water, contents of compartment, etc. As in pocketing, when surface permeability is neglected, the calculated results indicate less stability than the ship actually possesses.

Load Lines

Registered cargo-carrying ships are subject to the rules concerning minimum freeboard set down by the International Convention on Load Lines in 1966. The marking of the minimum allowable freeboard is painted on both sides of the ship near amidships. It is intended as a visual check of the required reserve buoyancy for each class of ship addressed by the load-line convention. Since the required minimum freeboard varies with water density (temperature and salt content) and severity of weather (location), different markings are used for minimum tropical fresh-water freeboard (*TF*), fresh-water freeboard (*F*), tropical-water freeboard (*T*), standard summer freeboard (*S*), winter freeboard (*W*), and the winter North Atlantic freeboard (*WNA*), in order of increasing required-minimum freeboard. The load-line

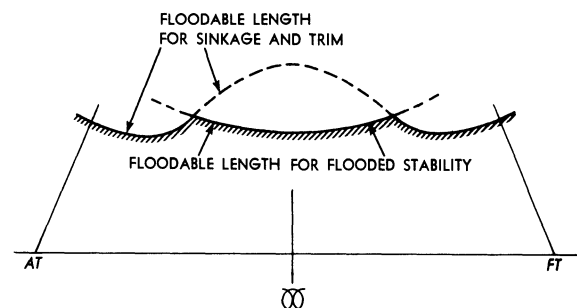


Figure 10-3. Floodable-length restraints

markings are illustrated in figure 10-4. A comprehensive discussion of load-line assignment, including domestic load lines for U.S. waters, is found in *Ship Design and Construction* (chapter 4).

10-2 Criteria for Subdivision of Naval Vessels

Numerous considerations are involved in determining the optimum arrangement of subdivisions for a naval

vessel, but the principal factors, according to a paper by Sarchin and Goldberg (1962) entitled "Stability and Buoyancy Criteria for U.S. Naval Surface Ships," are

1. Ability to survive underwater attack and damage
2. Protection of vital spaces against flooding
3. Interference of subdivision with arrangements
4. Interference of subdivision with access and systems

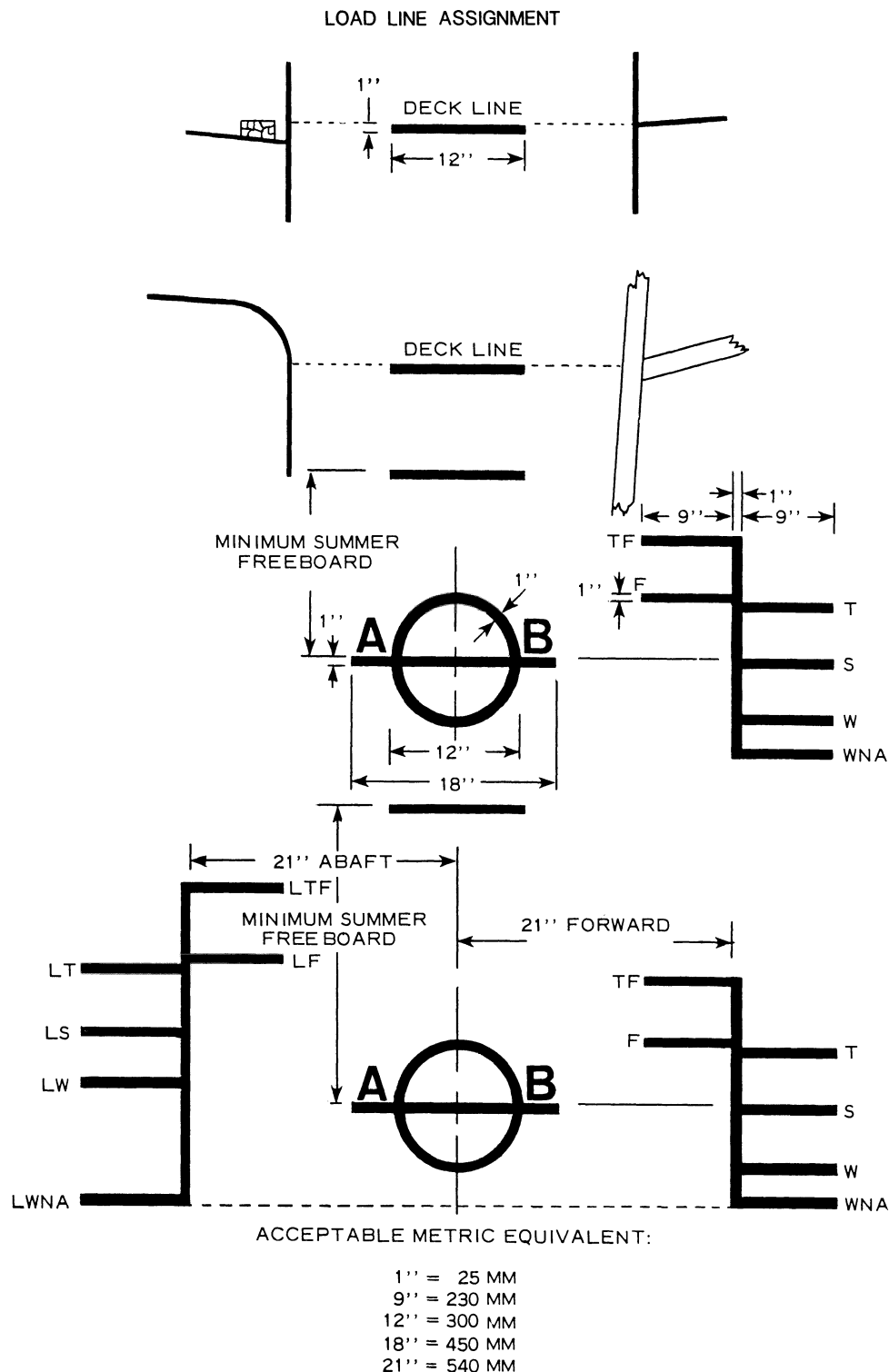


Figure 10-4. Load-line markings. (From Taggart 1980.)

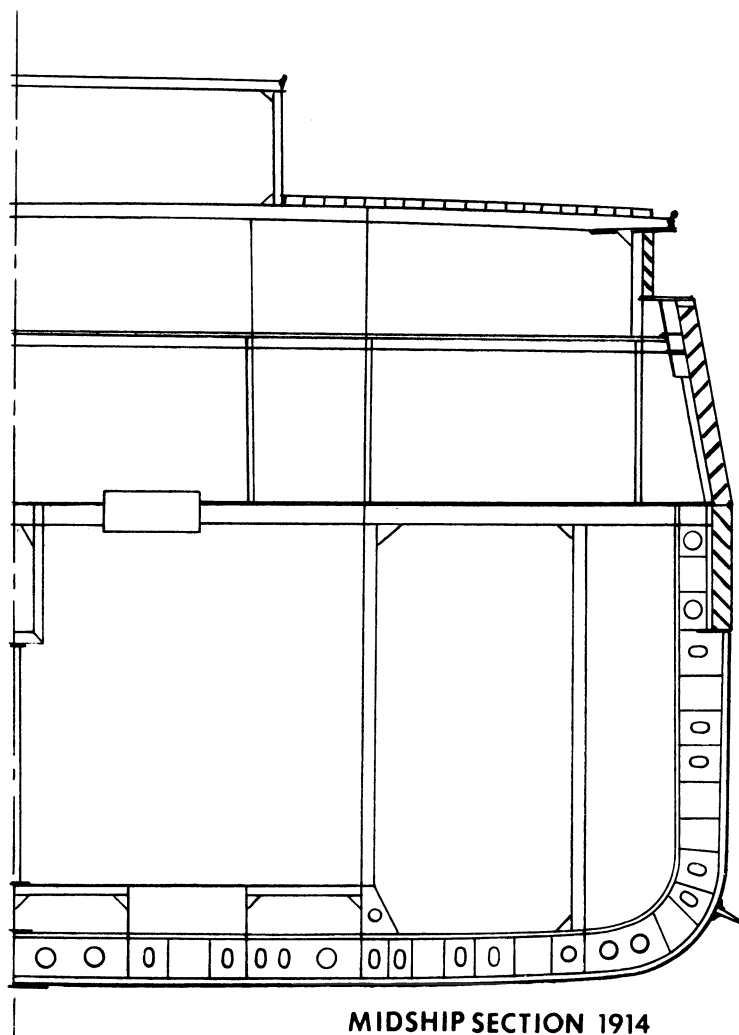


Figure 10-5a. Original side protection armor on the USS *Texas* (BB 35). (Drawing by Robert Sumrall.)

5. Provision for carrying liquids
6. Possibility of bow-collision damage
7. Possibility of stranding (section 10-3)

There are always conflicts among these various factors, and their relative importance must be determined.

ABILITY TO SURVIVE UNDERWATER ATTACK AND DAMAGE

In the middle of the nineteenth century, armor cladding was introduced into warship design, which started a competition between those who designed protective-armor systems for ships and those who designed the weapons that attempted to penetrate this armor. As the weapons became more powerful, the thickness of the armor increased, until the weight of the armor grew excessive. In order to reduce the weight of armor required, its use was restricted to protecting vital areas, such as the gun turrets, magazines, and machinery spaces that were sometimes grouped in a "citadel" (Breyer 1973). The introduction of torpedoes and mines in the 1870s greatly increased the possibility of sinking a war-

ship without first having to put its major weapons out of action and/or penetrating its above-waterline armor belt. The armor belt, which extended only slightly below the waterline, provided little protection from the damage inflicted by powerful torpedoes and mines. Around the turn of the century, the French Navy introduced the first fore-and-aft (longitudinal) "torpedo bulkheads," made of tougher nickel steel, which were designed to absorb the pressure waves and splinters of a torpedo hit. The coal bunkers on large warships were also located alongside the machinery spaces to provide additional energy absorption. Around 1910, the Austrian Navy introduced the armor-plated double bottom as a protection from explosions beneath the keel.

Large Combatants

During World War I, German warships proved to be much more resistant to underwater damage than those of the Allied navies, largely because of the superior watertight subdivision of the German ships. This experience led to the development of better torpedo-bulk-

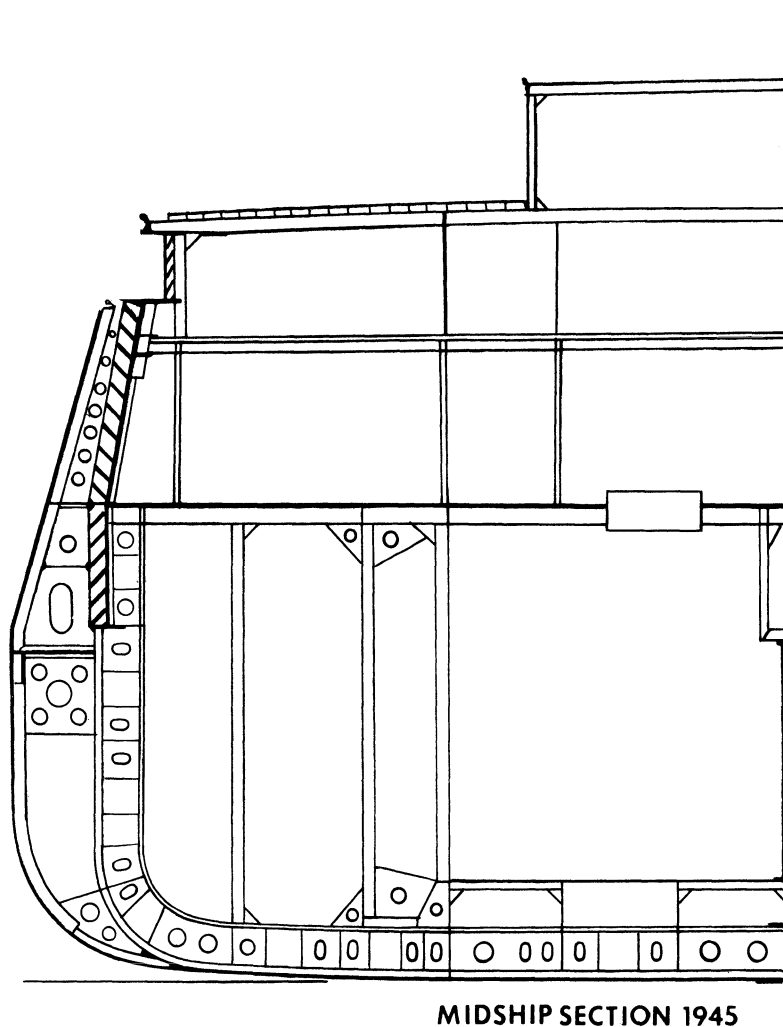


Figure 10-5b. Modifications to torpedo protection system on the USS *Texas* (BB 35). (Drawing by Robert Sumrall.)

head protection systems between the wars. An example of the conversions made to existing U.S. warships is shown in figures 10-5a and 10-5b. The U.S.S. *Texas* (BB 35) was built in 1914 and underwent a major overhaul in the 1920s. The original coal-fired boilers were replaced with smaller oil-fired boilers. With the space saved, the single-armored torpedo bulkhead was replaced with two torpedo bulkheads plus a "torpedo bulge" exterior to the original hull. Triple bottoms were added under the machinery spaces.

The U.S.S. *West Virginia* (BB 48) was completed in 1923. Her torpedo protection system (figure 10-6a) showed the lessons learned during World War I concerning resistance to underwater damage. The deep-wing tank closest to the skin of the ship was usually a void (filled with air) in which the explosion gases could expand, assuming a surface contact or proximity detonation. This was followed by two tanks filled with liquids (service or contaminated fuel oil or ballast) to absorb the energy of the blast and then by two additional voids to prevent the explosion from reaching the machinery

spaces. The protection system worked as designed (figure 10-6b) when the *West Virginia* was attacked by aerial torpedo bombers at Pearl Harbor in 1941. However, the magnitude of the damage suffered from seven torpedo hits was far beyond the design criteria for sustainable damage, and the ship sank on an even keel in shallow water. When salvaged and refitted, torpedo bulges were added, as they were to the other two battleships in the *Maryland* class in 1942.

Side-protective subdivision systems, incorporating four armored torpedo bulkheads and a triple bottom, were used in the last of the U.S. battleship classes constructed during World War II. Figure 10-7(a) illustrates the machinery-space subdivision used on the *Iowa*-class battleships (Dulin and Garzka 1976). This technique was expanded to five protective spaces on the *Essex*-class aircraft carriers, of which twenty-six were constructed during the war (figure 10-7(b)). The deep-wing tanks were further subdivided by numerous transverse bulkheads in order to contain and confine explosion damage to as small an area as possible. Further development in

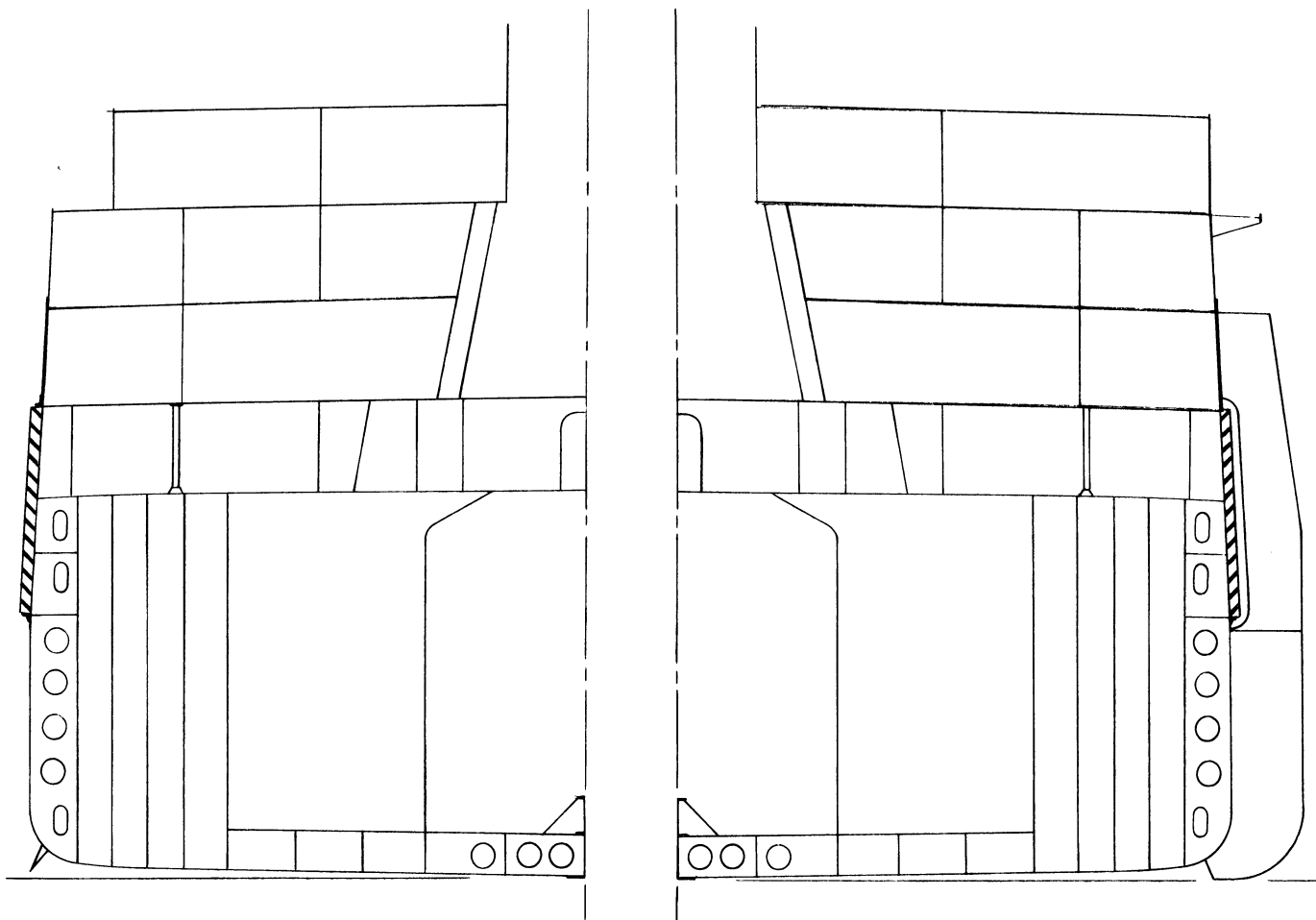


Figure 10-6a. Original and modified torpedo protection systems on the USS *West Virginia* (BB 48). (Drawing by Robert Sumrall.)

side protective systems for aircraft carriers has taken place since the war, with experimental tests or various tank configurations for both side and bottom protection. The changing nature of weaponry has also modified the design criteria for the ability to survive underwater attack.

Small Combatants

Because of the volume required for effective torpedo and mine protective systems, it is not possible to include them in the design of smaller warships. Destroyers, frigates, and small cruisers must depend on transverse bulkheads for adequate reserve buoyancy in order to survive torpedo damage. Longitudinal bulkheads generally have an unfavorable effect on damage stability, unless the off-center spaces formed by these bulkheads are cross connected to ensure rapid cross flooding, or unless these spaces are kept full of liquids. Single longitudinal bulkheads located on the centerline must be avoided, but carefully chosen and compartmented longitudinal bulkheads can reduce the amount of flooding

water that enters the ship while keeping list to a small angle. (This will be shown in an example problem.) Transverse bulkheads are normally carried watertight to a deck called the bulkhead deck, which coincides with the weather deck on most designs. The bulkhead deck may be stepped in the case of some destroyer types.

Decks and platforms other than the weather deck may have either a favorable or an unfavorable effect on transverse stability. If damage occurs below a watertight deck and the space below the deck floods completely, the effect is usually favorable (assuming the loss of buoyancy is not critical), because high flooding is prevented and free surface of flooding water is eliminated. On the other hand, if damage occurs above a watertight deck and flooding of spaces below is prevented, the effect of the watertight deck is unfavorable, since the ballasting effect of the low flooding will not be obtained. Because of the uncertainty as to the location of the damage relative to the deck and the probability that all decks will be ruptured except on the

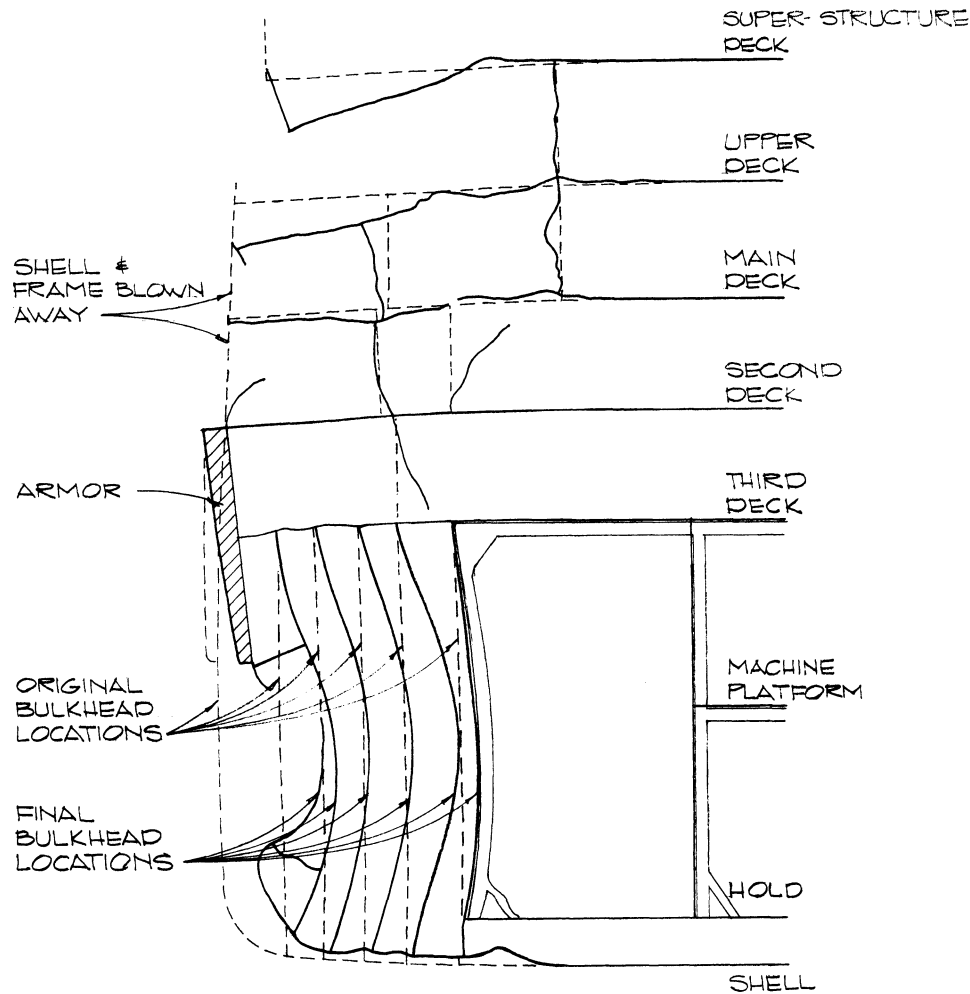


Figure 10-6b. Sketch of torpedo damage at frame 77 suffered by the USS *West Virginia* at Pearl Harbor

largest ships, one cannot rely on watertight decks and platforms below the weather deck in evaluating a ship's resistance to underwater attack. A watertight weather deck throughout the ship's length is desirable to prevent flooding of undamaged spaces, in the event of underwater damage that involves sufficient heel to submerge this deck.

PROTECTION OF VITAL SPACES AGAINST FLOODING

Vital spaces are defined as those spaces that are manned at general quarters and those unmanned spaces that contain equipment essential to the primary mission of the ship. It is obviously desirable to surround each of these spaces in the hull with a completely watertight envelope, since such protection might prevent flooding of the space, in the event of damage in the vicinity, and thereby preserve the function that the space accommodates.

The subdivision required for the protection of vital spaces will frequently involve above-waterline decks, whose effect was just discussed, and longitudinal bulk-

heads that may introduce unsymmetrical flooding exterior to the vital spaces. It is necessary to provide sufficient transverse stability to overcome these adverse effects by minimizing the resulting list or by providing for cross connections or counterflooding. When possible, the vital spaces should be located so as to avoid unsymmetrical flooding of the protective subdivision.

INTERFERENCE OF SUBDIVISION WITH ARRANGEMENTS

The provision of a series of transverse watertight bulkheads, necessary to develop resistance to underwater attack, will often interfere with obtaining the most favorable arrangement of spaces. Since all of the main transverse bulkheads extend continuously from the keel up, all compartments on the various levels between two main transverse bulkheads will usually be of the same length, whereas the optimum arrangement might require compartments of different lengths. The most favorable location of bulkheads, from the standpoint of resistance to underwater attack, may make it impossible to obtain the desired length of main compartments,

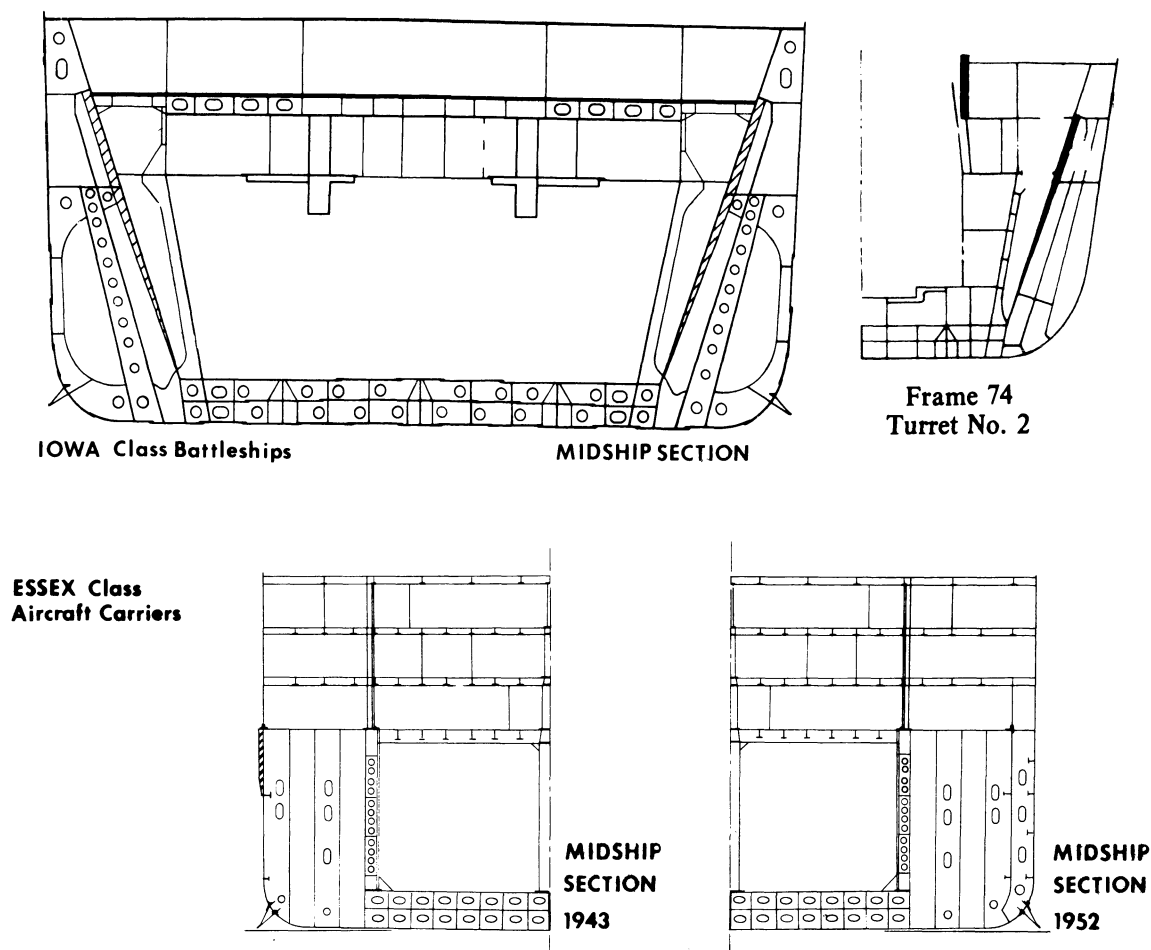


Figure 10-7. Side protection systems for large combatants during World War II. (Drawing by Robert Sumrall.)

from the standpoint of arrangement, because of the floodable length. Generally, the desired locations for the machinery spaces and magazines establish the approximate location of the adjacent main transverse bulkheads. In addition to placing bulkheads to satisfy floodable-length criteria, they may also be required to satisfy transverse stability considerations. This can further complicate internal arrangements.

INTERFERENCE OF SUBDIVISION WITH ACCESS AND SYSTEMS

Penetration of watertight subdivisions by piping, electric cables, ventilation ducts, and access openings involves considerable weight and expense, since watertight fittings must be provided. An additional disadvantage is that the access and ventilation closures must be set, throughout the life of the ship, in accordance with the various damage-control doctrines, and rapid access is hindered by the necessity for opening and securing doors in the process of passing through the subdivisions.

The bulkheads with the greatest number of penetrations, and through which rapid access is most often required, are those between the main and second decks in the midship region. If these bulkheads are not wa-

tertight, it must be assumed that for cases of damage in the region of the nontight bulkheads, flooding will occur fore and aft along the second deck until watertight bulkheads are reached. This is equivalent to disregarding the buoyant effects of the portion of the ship above the second deck between the watertight bulkheads. On some ships having a relatively high freeboard, damaged-stability investigations may show that damaged stability and reserve buoyancy will be adequate, with some of the main transverse bulkheads considered nontight above the second deck. In such a case, it is essential to make the second deck watertight to avoid progressive flooding into the spaces below.

In cases where it is necessary to take full advantage of all buoyant volume below the weather deck in order to meet the criteria for resistance to underwater damage, it will usually be possible to accept some nontight penetrations of the main transverse bulkheads without introducing any appreciable danger of progressive flooding into intact spaces. The procedure for establishing the areas in which nontight penetrations are acceptable is described below and illustrated by figures 10-8 and 10-9.

For any bulkhead, such as bulkhead A in figure 10-8, the designer determines the maximum extent of flood-

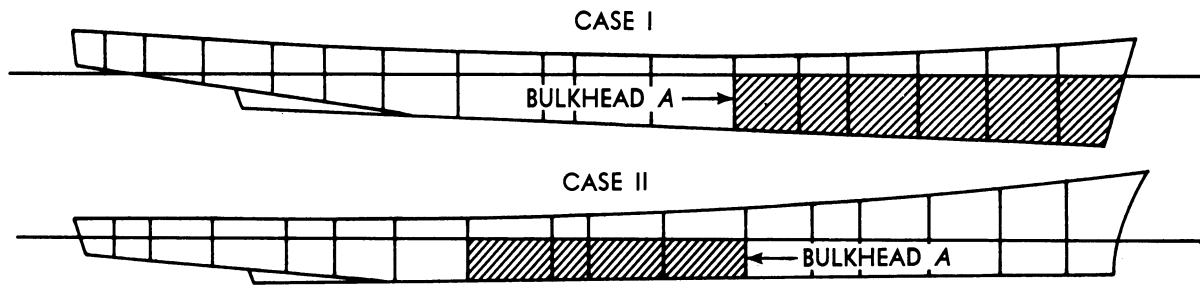


Figure 10-8. Limitations to compartmentation

ing of the adjacent compartments that can occur both immediately forward and immediately aft of bulkhead A without causing the ship to capsize or founder. These extents of flooding are represented by the cross-hatched portions of the ship in figure 10-8.

The designer will next calculate the drafts forward and aft for each of these two cases of flooding and draw the trimmed waterlines at which the ship will float in each case.

He now determines which of the two cases of flooding will produce the deeper immersion of bulkhead A. This establishes the maximum height to which the water will rise on bulkhead A, assuming the ship settles without heeling in still water, for the two most extensive cases of flooding of adjacent compartments that the ship can survive and that could involve danger of progressive flooding through bulkhead A.

Allowances for heel due to symmetrical flooding, roll, and wave action are applied in the manner illustrated by figure 10-9. This illustration represents bulkhead A of figure 10-8; point A corresponds to the intersection of the deeper trimmed waterline of figure 10-8 (case 1) with bulkhead A.

It is assumed that the ship is subject to an angle of heel of 15° . It is also assumed that the ship is rolling to an angle ϕ , which is a function of ship type and is diagrammed in figure 10-9. Waterline BC is drawn through point A to represent the condition with the ship inclined to an angle equal to 15° plus ϕ . To allow for wave action, line DE is drawn 4 feet above waterline BC. Line DE and the corresponding line on the opposite side of the ship, which are designated as the V-lines for bulkhead A, outline the area FGH through which nontight penetrations are acceptable.

The weather deck should, of course, be watertight to prevent flooding from the sea when the ship is heeled over as the result of underwater damage. Similar tightness of the second deck should be provided in regions where transverse bulkheads between the second and main decks are nontight. As indicated above in the case of the transverse bulkheads, there may be an area near the ship's centerline in which nonwatertight penetrations are acceptable. The extent of this area is determined in a manner similar to that used in establishing

the V-lines for the transverse bulkheads. Where ventilation penetrations occur outboard of the V-lines for the deck, the penetration may be made watertight either by installing a watertight closure at the deck or by making the ventilation duct watertight up to its intersection with the V-line.

PROVISION FOR CARRYING LIQUIDS

In naval vessels the location of liquid loading has traditionally been important, probably more so prior to our era of missiles and atomic weapons. However, deep side tanks are still the most effective protection from underwater side attack for large warships. It must be understood that because of their large transverse moment and the possibility of free-surface effects, deep side tanks may have a detrimental effect on transverse stability. This disadvantage can be reduced and controlled by the installation of cross-connecting pipelines with control valves. Double-bottom tanks are generally preferable and are more commonly used in both warships and merchant ships. They not only provide protection from underwater damage but form an integral part of the ballasting system and center-of-gravity control both vertically and longitudinally. Since inner bottom tanks are often partially filled with liquids, they

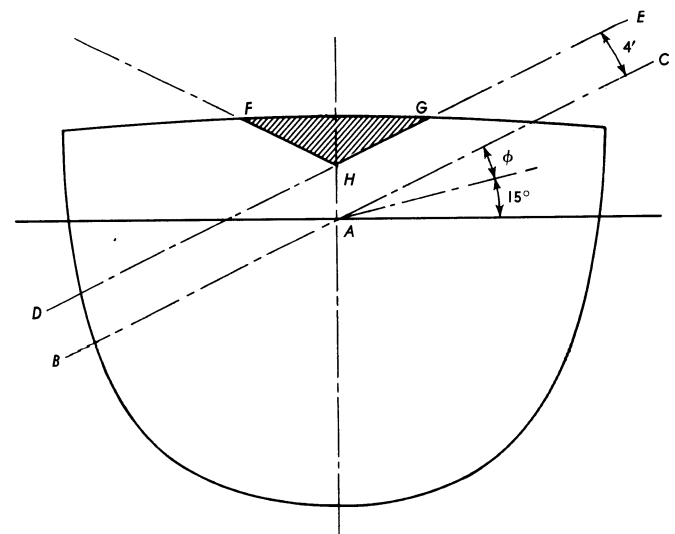


Figure 10-9. Limitations for deck openings

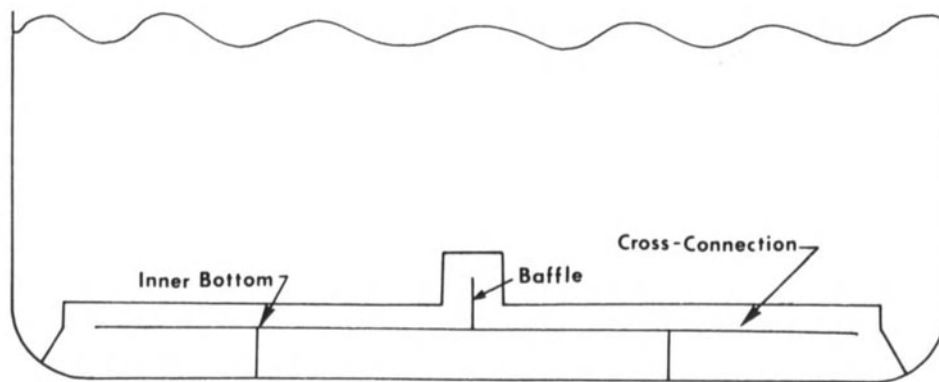


Figure 10-10. Cross connection for double-bottom tanks

should be segmented by baffles to reduce free-surface effects and be cross connected to prevent unsymmetrical flooding when damaged (figure 10-10). Their effect in grounding and stranding will be discussed later.

BOW COLLISION DAMAGE

Collisions involving bow damage are seldom fatal to the ship, because flooding can be minimized by locating a watertight transverse bulkhead at least 5 percent of the

ship's waterline length abaft the forward perpendicular. This is generally used in merchant ships and small craft as well. It is called the *collision bulkhead*.

10-3 Grounding and Stranding

Historically, thousands of seagoing vessels have been lost through grounding and stranding because of navigational errors, loss of propulsion, storms, and poor seamanship. Ships are designed to accept the structural loads associated with a severe storm on the open ocean, but most ships grounded on rocks or a coral reef cannot withstand the structural loads caused by pounding of the surf. Some smaller vessels that operate in regions of large tidal variations are designed to withstand intentional grounding and have hull forms that are stable, regardless of the depth of water (see section 6-6). Sailing vessels with considerable deadrise are unstable when aground, but flat-bottomed and low-deadrise ships have adequate transverse stability when grounded on an even bottom (figure 10-11), even if $\overline{GM}$ goes negative (equation 6-24).

If aground only on a narrow portion of the ship such as the bow (figure 10-12), the ship will heel as the tide goes out, until the heeling moment set up by the ground reaction equals the righting moment from the ship's static-stability curves (adjusted for trim). In general, a stranded ship will not capsize unless the angle of incli-



Figure 10-11. The SS *Ellin* aground off Point Arguello, California

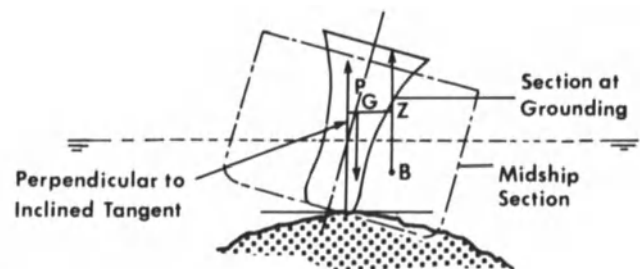


Figure 10-12. Ship grounded at the bow

nation approaches the range of positive stability when afloat, as corrected for the apparent rise in the $\overline{KG}$ using equation 6-23.

Broaching is a term used to describe the situation in which a ship veers broadside to the wind and waves. When the ship is grounded, broaching refers to being driven parallel to shore. Broaching should be avoided by laying anchors to seaward, if circumstances permit. Once a stranded ship has broached, it is generally driven further aground by wave action acting along the entire length of the ship. A broached ship is also subject to capsizing by breaking waves.

The most effective subdivision for protection against flooding, if grounding has damaged the bottom plating, consists of a complete inner bottom. (In smaller vessels, where it is often impractical to fit an inner bottom, it may be possible to make the lowest platform deck watertight.) The double-bottom spaces should be subdivided as extensively as possible. This increases the flexibility of their usage in the ballasting systems and generally does not interfere with the other subdivision criteria.

From the stability standpoint, when stranding without damage to the watertight integrity of the hull occurs, there are several interesting problems involving the center of gravity, one of which was pointed out in chapter 6 under docking. The other involves the action of removing weight from the ship in order to refloat her. The process, called *jettisoning*, should be considered when necessary.

JETTISONING WHEN AGROUND

In a ship floating freely, jettisoning is merely a problem of weight removal. With the grounded ship, jettisoning introduces several possibilities. It should be noted first, however, that from the standpoint of good seamanship and the overall success of refloating a grounded ship, jettisoning is not considered good practice when the lightened ship cannot be prevented from going further aground. The decision to jettison, of course, should be made by the commanding officer, depending upon the circumstances. Where the danger of further grounding does not exist or is remote, such as in a case of grounding on a mud flat in protected water, there is often no reason why jettisoning should not be a practical course of action if other less expensive methods have failed.

In originating an equation for jettisoning a weight when aground (assuming the ship does not refloat), one approach is to consider the problem in light of two separate situations: weight removal and grounding.

Let us establish that a ship aground, having jettisoned a weight, is in the same condition from the stability standpoint as the same ship that jettisoned the same weight from the same place while afloat and then grounded to the same draft.

With the second condition in mind, let

- Δ = original displacement afloat
- $\overline{KG}$ = original height of G afloat
- w = weight jettisoned from $\overline{Kg}$
- F_B = grounded displacement (by draft)
- $\overline{KG}_2$ = final virtual height of G
- $\overline{KG}_A$ = height of G afloat after weight removal
- Δ_A = displacement after weight removal.

Then

$$\overline{KG}_A = \frac{\Delta \overline{KG} - w \overline{Kg}}{\Delta - w} \quad (10-3)$$

Now the ship goes around and, from the “docking” equation (6-23),

$$\overline{KG}_{\text{Dock}} = \frac{\Delta_A \overline{KG}_A}{F_B}$$

we may substitute the terms above so that

$$\overline{KG}_2 = \frac{(\Delta - w) \frac{\Delta \overline{KG} - w \overline{Kg}}{\Delta - w}}{F_B}$$

or

$$\overline{KG}_2 = \frac{\Delta \overline{KG} - w \overline{Kg}}{F_B} \quad (10-4)$$

where G_2 is the virtual position of the center of gravity of a grounded ship, considering the jettisoned weight. Here, F_B is still the original displacement minus the upward ground force, or simply, the final displacement or buoyant force at the grounded draft. It is the same as before jettisoning, because both the original displacement and the upward force due to grounding are reduced by the amount of the jettisoned weight.

ACTIONS WHEN HARD AGROUND

For practical considerations of safety, the addition of low weights to the ship is often the best approach. When hard aground on an unprotected reef, beach, or lee shore, the best action is to flood low compartments and wait for assistance. Ships equipped with main propulsion steam condensers must avoid excessive “backing down,” since sand or mud may be ingested from the propeller wash during astern operation.

The use of salvage tugs equipped with “beach gear” capable of 60 tons of pull per set is discussed in books on ship salvage, including the *U.S. Navy Ship Salvage Manual*. Figure 10-13 shows one of the efforts to pull the U.S.S. *Missouri* (BB 63) off Thimble Shoals in Hampton Roads in 1950. The combination of salvage vessels and beach gear that successfully refloated the



Figure 10-13. Pulling the USS *Missouri* free

Missouri in February of 1950 is diagrammed in figure 10-14.

Before pulling off a stranded vessel, the hull must be examined very carefully for structural damage and signs of flooding. Pulling off a seriously weakened ship could result in her loss from progressive flooding. The successful attempt to salvage the seriously damaged U.S.S. *Frank Knox* (DDR 742) following grounding on an uncharted reef is discussed in *Shipboard Damage Control* (Bissel *et al.* 1976, chapter 6).

10-4 Structural Impairment and Shock Phenomena

Aside from damage that penetrates a ship's hull and invades its watertight integrity, there are other types of underwater, noncontact damage (particularly to naval vessels) that must be considered in the design. Where damage anticipated from contact invasions of the hull is of a localized nature and the design action is toward

containment, the shocks produced by underwater explosions will generally extend the damage. Structural failure will be observed over a wide area, machinery and other heavy equipment will be displaced from their foundations, and a general reduction in longitudinal strength may take place. It is important to be aware of the nature and origin of this type of hazard.

When a shock motion is introduced by the action of underwater explosion, a wide variety of vibrational motions may result in various parts of the ship, as shown in figure 10-15. The maximum accelerations occur at high frequencies in the lower decks near the explosion. As the vibration wave travels throughout the ship, there is a shift in the peak of the shock spectra to lower frequencies. In the upper decks, a subaudible peak occurs in the spectra, indicating potentially large low-frequency motions. The characteristic frequencies of underwater shocks should be studied by the naval ship designer in selecting types of equipment and equipment mounts for location in various portions of the ship. Extensive literature and test data are available from the David W. Taylor Naval Ship Research and Development Center for use in analyzing structural vibrations induced by shock phenomena. These data can be compared with the ship's natural vibration characteristics to determine if a given shock spectrum will induce resonant vibrations in the structure.

A commonly used design requirement for shock analysis is the *keel shock factor*, which is a measure of the shock severity at the keel and is given for a naval ship in the particular general specifications. There are many ways to design for shock resistance. One method is to "harden up" or increase the ruggedness of components and systems intended to resist shock motions. Components are tested on medium-weight shock machines or on floating test barges subjected to controlled explosions in order to evaluate their ability to resist shock. Another approach is to attenuate the shock wave by selecting either hard mounts or resilient mounts in accordance with the expected shock spectra (that is, hard mounts for low frequencies and resilient mounts for high-frequency attenuation, when the equipment-mounting requirements permit such a solution).

10-5 Weight Control on Naval Vessels

The seaworthiness of ships and the success of damage control depend mainly on careful weight control. Two important parameters of a ship affecting her damaged stability are her intact reserve buoyancy and center of gravity. Since the weight of the ship and her distributions change during her lifetime, a close watch of the weight alterations is necessary.

Each ship is required to have a certain amount of reserve buoyancy and minimum values of parameters

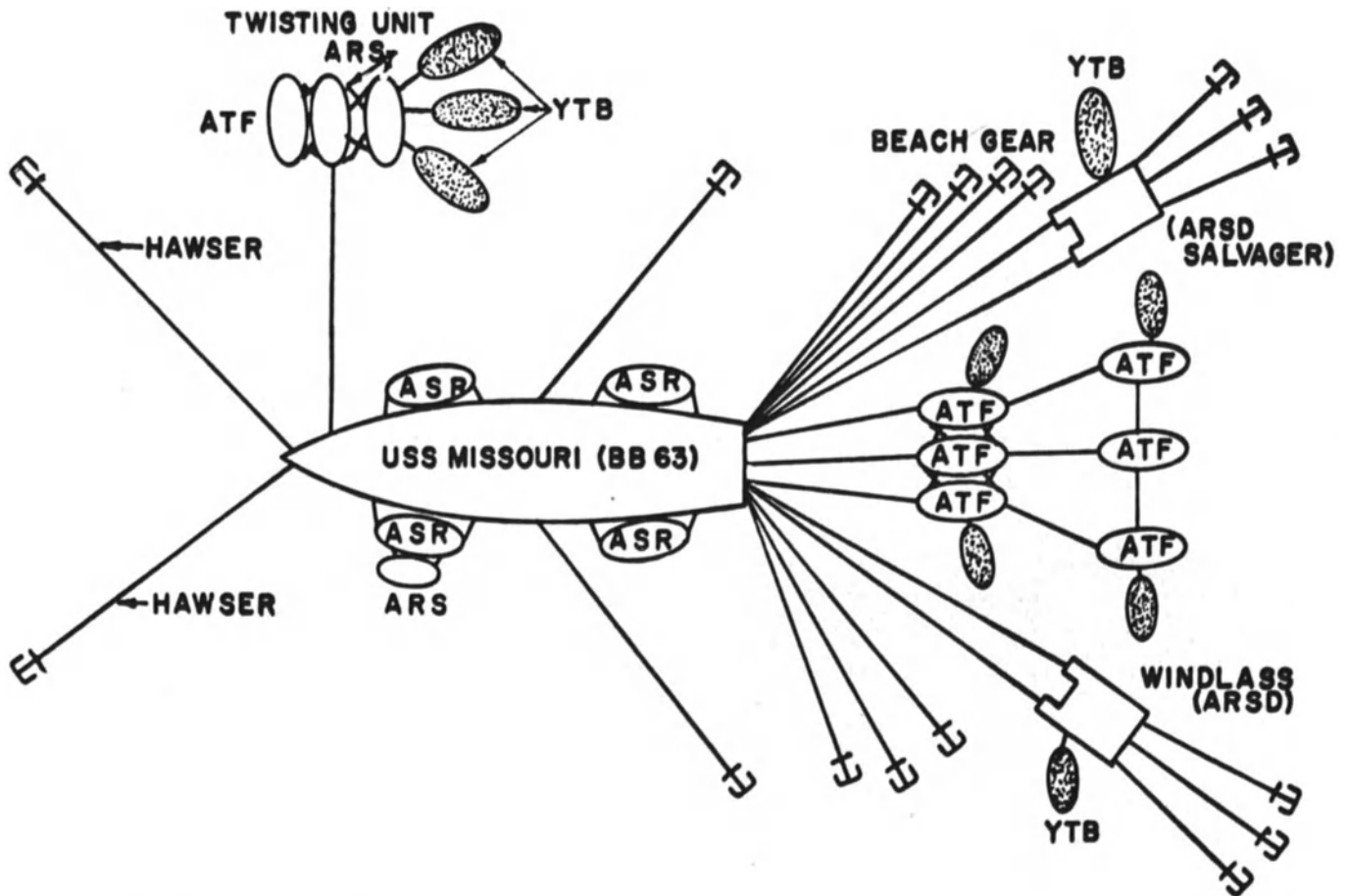


Figure 10-14. Pulling the USS *Missouri* free

concerning the righting-arm curve (section 8-9). These two requirements can be translated in terms of the maximum displacement and the maximum $\overline{KG}$ a ship is allowed to have. Depending on the present weight and $\overline{KG}$ of a naval ship, a status is assigned. Since there are two main variables, four combinations are possible.

Status 1—A reasonable increase in weight and a rise of the ship's center of gravity are acceptable. Added weight and moment resulting from changes, if not excessive, will not require any compensation for the present.

Status 2—Neither increase in weight nor rise of the ship's center of gravity can be accepted. Compensation for adding a high weight may be obtained by removing an equivalent weight from a location at least as high.

Status 3—A reasonable increase in the ship's weight is acceptable, but a rise of the ship's center of gravity must be avoided. Compensation can be obtained by adding ballast, removing a high weight, etc.

Status 4—A reasonable rise of the ship's center of gravity is acceptable, but increase in weight must be avoided. Compensation for added weight may be obtained by removal of an equal or greater weight at any level.

The addition of weight itself, even low in the ship, must be avoided wherever possible in tightly designed combatant ships. It takes only 20 to 40 tons to increase the draft of a destroyer-type ship one inch. The resulting additional displacement adversely affects the ship's ability to make required speed and decreases its reserve buoyancy. It also alters the stability characteristics from those that have been previously computed.

Once the planned weight additions for a new system are known, it is possible to compensate for their effect by removing a weight elsewhere. This *weight and moment compensation technique* involves the straightforward use of equation 6-9 or 6-10.

The Naval Ship Systems Command administers the weight and moment compensation program. Each approved SHIPALT or allowance-list change indicates the magnitude of the weight and moment changes and earmarks the specific items to be used as compensation. In certain cases, where compensation is not available but the item to be added has important military value, the Command may authorize installation without compensation. Such a decision is reached after the Chief of Naval Operations has been advised of the resultant reduction in ship performance (speed, stability, etc.).

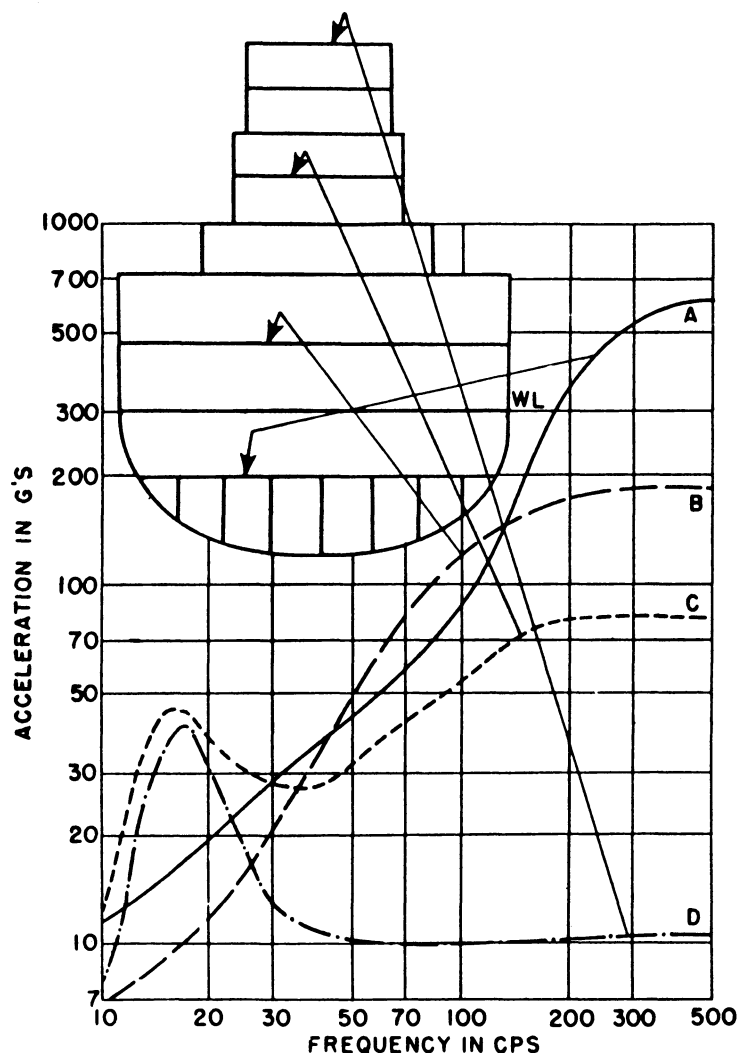


Figure 10-15. Typical shock spectra at different levels in a ship. (Courtesy of Harry L. Rich.)

10-6 Free Communication with the Sea

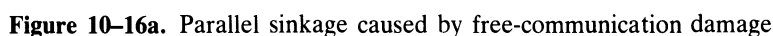
When one or more of the exterior boundaries of a ship are ruptured, so that the sea may flow freely into and out of the damaged compartment with a minimum of restrictions as the ship rolls, the condition is described as partial flooding with *free communication* with the sea. Under this condition, $\overline{GZ}$ and $\overline{GM}$ may be initially increased because of the lowering in G due to the added weight of flooding water. (Since flooding is due to gravity, the floodwater is below the waterline, and in most surface ships the center of gravity of the flooded water is below the ship's center of gravity.) They are reduced subsequently because of the virtual rise in G , due to the free-surface effect of the flooding water and the so-called free-communication effect. This latter effect, illustrated in figure 10-16, occurs only when the partially flooded compartment is not centered on the centerline of the ship.

In the case illustrated, the off-center compartment shown is flooded through a hole in the exterior bound-

ary of the ship. (Note that the compartment is vented above.) Because of the added weight of the flooding water, the ship will sink to the new waterline W_1L_1 . This will cause a change in the position of G and, because of the increased displacement, will relocate M , establishing a new static-stability curve that must be corrected for the final position of G .

With the off-center compartment flooded, the ship will assume a list that is further aggravated by the free-surface effect of the flooding water. As the ship lists, additional water will flow into the compartment, tending to level off at the height of the external waterline. The list will increase because of this tendency, causing still more water to enter, and in turn causing greater list in decreasing increments, until a final list corresponding to the waterline W_2L_2 is reached.

This last cause of list, beyond that brought on by the effect of the initial off-center flooding water and its free-surface effect, is due to the increasing amount of loose water flowing into the off-center compartment. This



An exact evaluation of the free-communication effect by the method studied thus far is virtually impossible due to the fluctuation in the amount of flooding water, and hence the ship's displacement, with the angle

The effect of the initial and overall stability of a ship, when free communication with the sea occurs, may be considered to take place in the following steps (see figures 10-16a and 10-16b):

1. A mass of water is admitted and causes the ship to sink parallel to its original waterline WL to W_1L_1 .



The amount of this water (w) is determined by the volume of the flooded compartment below W_1L_1 . There is a vertical shift of the ship's center of gravity from G_0 to G_1 , owing to this added weight (usually downward).

$$\overline{KG}_1 = \frac{\Delta_0 \overline{KG}_0 + w \overline{Kg}}{\Delta_1}$$

where Δ_0 = original displacement (at waterline WL)
 w = weight of flooding water up to waterline W_1L_1
 $\Delta_1 = \Delta + w$ = displacement of the ship at waterline W_1L_1
 $\overline{KG}_0$ = original vertical height of the ship's center of gravity
 $\overline{KG}_1$ = vertical height of the ship's center of gravity when displacing Δ_1 tons
 $\overline{Kg}$ = vertical height of the center of gravity of the floodwater w .

2. A free surface is created that causes a virtual rise in the center of gravity from G_1 to G_2 .

$$\overline{G_1G_2}(\text{vert}) = \frac{i}{\nabla_1}$$

where ∇_1 = volume of displacement at waterline W_1L_1
 i = moment of inertia of the free-surface area about a longitudinal axis through its centroid.

3. The ship lists due to the off-center weight w and causes an additional quantity of water to flow in. Note that if the shell opening did not exist, the ship would list to the dotted waterline and the surface of the floodwater would parallel the dotted waterline along line ef . The *additional flooding water* w_1 , represented by the area $efkl$, is dependent upon the horizontal cross-sectional area of the bilged compartment, the angle of list, and the distance from the center of gravity of the compartment to the ship's centerline. The addition to displacement caused by w_1 will be ignored as a second-order effect.

$$w_1 = a(y \tan \phi)/35 \text{ ft}^3/\text{ton} \quad (10-5)$$

where a = the surface area

y = the distance to the centroid of the flooding water in the upright condition.

In order to calculate the total effect on $\overline{GM}$ and the angle of list, it is easier, geometrically, to sum moments about the original center of gravity of the ship, as illustrated in figure 10-17, which is an expanded view of figure 10-16b.

The inclining moment of the off-center weights about the original center of gravity G_0 relative to the new

waterline W_2L_2 is

$$\begin{aligned} \text{inclining moment} &= w(y \cos \phi - \overline{G_0g} \sin \phi) \\ &+ w_1(y \cos \phi - \overline{G_0g_1} \sin \phi) \end{aligned} \quad (10-6)$$

where

$$\overline{G_0g_1} = \overline{G_0W_1L_1} - y \tan \frac{\phi}{2}$$

Combining equations 10-5 and 10-6,

$$\begin{aligned} \text{moment} &= wy \cos \phi - w \overline{G_0g} \sin \phi \\ &+ \left(\frac{ay^2}{35} \tan \phi \right) \cos \phi \\ &- \left(\frac{ay}{35} \overline{G_0g_1} \tan \phi \right) \sin \phi \\ &= wy \cos \phi + \left[\frac{ay^2}{35} \left(1 - \frac{\overline{G_0g_1}}{y} \tan \phi \right) \right. \\ &\quad \left. - w \overline{G_0g} \right] \sin \phi \end{aligned} \quad (10-7)$$

However, $\frac{\overline{G_0g_1}}{y} \tan \phi$ will normally be a very small quantity, since G_0 and g_1 will both be near the waterline W_1L_1 , and the analysis is *limited to small angles* so that M_1 can be defined. In other words, the correction for the vertical location of w_1 will be neglected. This leaves

$$\begin{aligned} \text{inclining moment} &= wy \cos \phi \\ &+ \left[\frac{ay^2}{35} - w \overline{G_0g} \right] \sin \phi \end{aligned} \quad (10-8)$$

The righting moment relative to G_0 is

$$\Delta_1 \overline{G_0Z} = \Delta_1 (\overline{G_0M_1} - FSC) \sin \phi \quad (10-9)$$

Equating these moments for the condition of static equilibrium,

$$\begin{aligned} wy \cos \phi + \left[\frac{ay^2}{35} - w \overline{G_0g} \right] \sin \phi \\ = \Delta_1 (\overline{G_0M_1} - FSC) \sin \phi \end{aligned} \quad (10-10)$$

Collecting the $\sin \phi$ terms and dividing by Δ_1 (note that ∇_1 equals $35\Delta_1$),

$$\frac{wy}{\Delta_1} \cos \phi = \left[\overline{G_0M_1} + \frac{w \overline{G_0g}}{\Delta_1} - FSC - \frac{ay^2}{35} \right] \sin \phi$$

Note that

$$\frac{w \overline{G_0g}}{\Delta_1} = \overline{G_0G_1}(\text{vert})$$

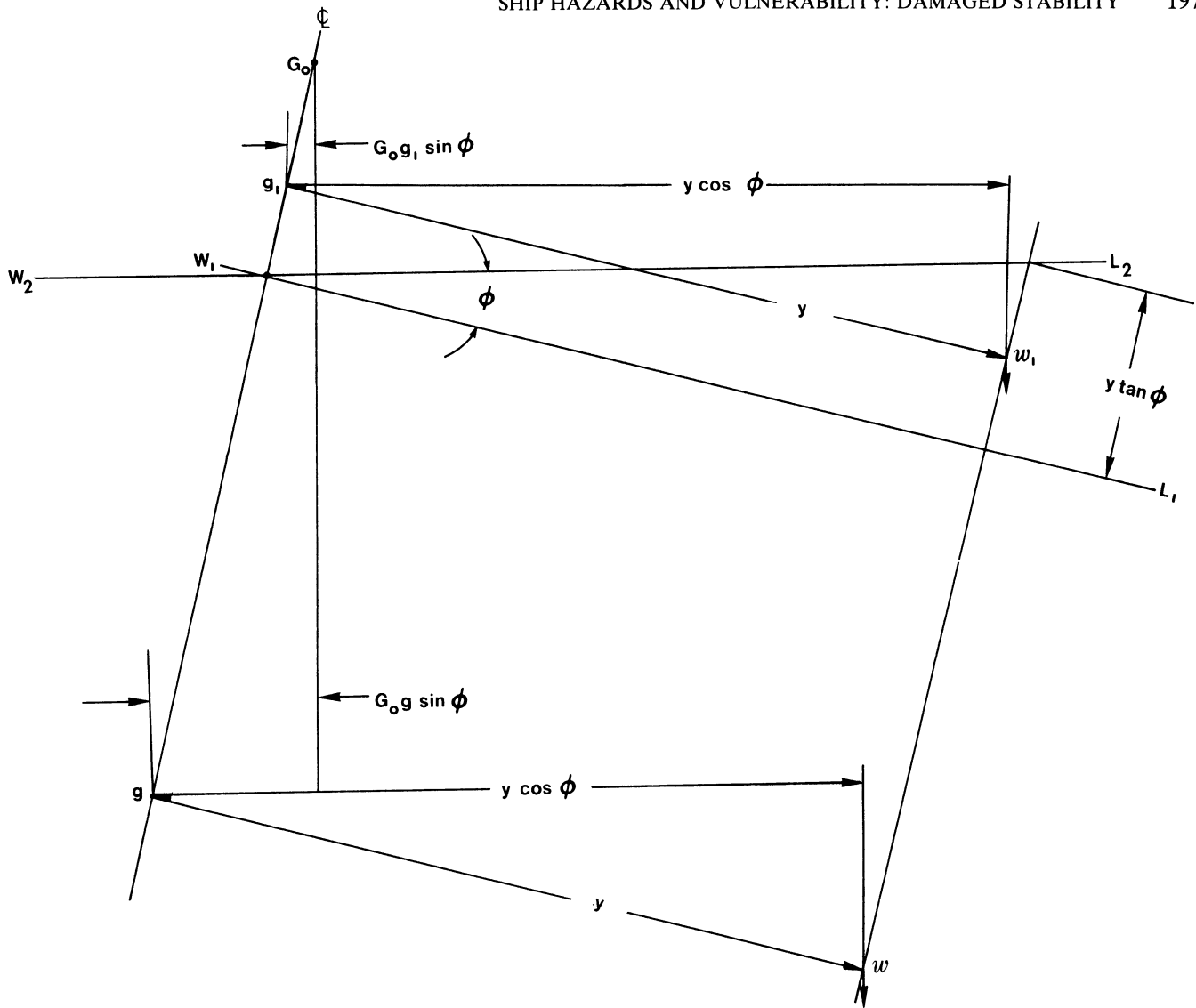


Figure 10-17. Expanded view of weight shifts in figure 10-16b

and

$$\overline{G_0 M_1} + \overline{G_0 G_1} (\text{vert}) = \overline{G_1 M_1}$$

Thus,

$$\frac{wy}{\Delta_1} = \left[\overline{G_1 M_1} - FSC - \frac{ay^2}{\nabla_1} \right] \tan \phi \quad (10-11)$$

The term $\frac{ay^2}{\nabla_1}$ represents a reduction in $\overline{GM}$ for *small angles of heel and will be called the free-communication correction (FCC)*. Like the free-surface correction, it represents a virtual rise in $\overline{G}$ since it is applied as a sine correction. An effective $\overline{GM}$ can be defined as

$$\overline{GM}_{\text{eff}} = \overline{G_1 M_1} - FSC - FCC \quad (10-12)$$

and the angles of list may be found from

$$\tan \phi = \frac{wy}{\Delta_1 \overline{GM}_{\text{eff}}} \quad (10-13)$$

provided $\phi < 10^\circ$

Note that the weight used in this calculation is the parallel sinkage weight addition w , and that the effect of the additional flooding weight w_1 is accounted for in $\overline{GM}_{\text{eff}}$.

Added Weight Versus Lost Buoyancy

In the foregoing discussion, it was assumed that when a ship undergoes flooding, either partial or solid, with or without free communication with the sea, the weight or displacement has been increased by the weight of the flooding water. This method, which may be called the *added-weight method*, assumes that none of the hull surface exposed to the buoyant forces of the water is lost.

The *lost-buoyancy method* (Comstock 1967, chapter 3, section 7) assumes that flooding water in free communication remains part of the sea, and therefore, the part of the hull that has been flooded no longer con-

tributes buoyancy to the ship. In other words, the vertical pressure forces about the flooded compartment act upon the sea rather than upon the ship—hence the name, *lost buoyancy*. Consider an empty box-shaped lighter, whose weight is entirely dependent upon its structure, divided longitudinally into three compartments. The center compartment is then flooded in free communication. By the lost-buoyancy method, this flooding water remains part of the sea, and the middle section of the lighter possesses no buoyancy. However, the structural weight of the lighter remains the same, whether flooded or not, and must be supported by a buoyant force equal to this weight. The lost buoyancy amidships must be made up at the buoyant end sections by increasing the draft such that the water displaced by the end sections equals the structural weight of the lighter. To sum up the foregoing, the added-weight method deals with a change in weight, whereas the lost-buoyancy method deals with a change in underwater shape.

One can itemize the effects of each method as follows:

1. Added-weight method causes
 - a) Change in tons displacement
 - b) Change in volume of displacement
 - c) Change in draft, trim, and list
 - d) Change in location of center of gravity
 - e) Change in location of center of buoyancy
 - f) Change in location of transverse and longitudinal metacenters
 - g) Need for free-surface and free-communication correction to effective metacentric height.
2. Lost-buoyancy method causes
 - a) No change in tons displacement
 - b) No change in volume of displacement
 - c) Change in draft, trim, and list
 - d) No change in center-of-gravity location
 - e) Change in center of buoyancy
 - f) Change in location of transverse and longitudinal metacenters
 - g) No free-surface or free-communication correction.

Example 10-1: Compare the list angles that result from free communication involving two adjacent compartments, each 35 feet long on rectangular barge *A* (which is subdivided by five transverse watertight bulkheads and two equally spaced longitudinal bulkheads). Rectangular barge *B* has the same transverse-bulkhead arrangement but has two longitudinal bulkheads, located one-fifth of the beam in from each side of the vessel. The barges viewed from above in figure 10-18 are 210 feet long, 60 feet wide, and draw 14 feet in the undamaged condition. Before damage the $\overline{KG}$ is equal to 11.5 feet.

Solution: The new mean drafts following damage are as follows:

Case A: Each subdivision volume equals $\left(\frac{L}{6}\right)\left(\frac{B}{3}\right)T$ and $\nabla_0 = \nabla'_A$.

$$18\left(\frac{L}{6}\right)\left(\frac{B}{3}\right)T_0 = 16\left(\frac{L}{6}\right)\left(\frac{B}{3}\right)T_A$$

$$T_A = \frac{18}{16}T_0 = \frac{18}{16}(14) = 15.75 \text{ ft}$$

Case B: Each subdivision volume equals $\left(\frac{L}{6}\right)\left(\frac{B}{5}\right)T$ and $\nabla_0 = \nabla'_B$.

$$30\left(\frac{L}{6}\right)\left(\frac{B}{5}\right)T_0 = 28\left(\frac{L}{6}\right)\left(\frac{B}{5}\right)T_B$$

$$T_B = \frac{30}{28}T_0 = \frac{30}{28}14 = 15 \text{ ft}$$

where ∇' = actual buoyant volume.

The weight of the flooding water w at the new draft is

$$w_A = \frac{2(35)(20)(15.75)}{35} = 630 \text{ tons}$$

$$w_B = \frac{2(35)(12)(15)}{35} = 360 \text{ tons}$$

The displacements before and after damage are

$$\Delta_0 = \frac{LBT_0}{35} = \frac{210(60)14}{35} = 5,040 \text{ tons}$$

$$\Delta_A = \Delta_0 + w_A = 5,670 \text{ tons}$$

$$\Delta_B = \Delta_0 + w_B = 5,400 \text{ tons}$$

Assume the $\overline{Kg}$ of the floodwater is equal to the draft divided by two.

$$\overline{KG}_A = \frac{\Delta_0 \overline{KG}_B + w \overline{Kg}}{\Delta_A}$$

$$= \frac{11.5(5,040) + 630(7.875)}{5,670}$$

$$= 11.1 \text{ ft}$$

$$\overline{KG}_B = \frac{11.5(5,040) + 360(7.5)}{5,400}$$

$$= 11.23 \text{ ft}$$

$$\overline{KM}_A = \frac{T_A}{2} + \frac{B^2}{12T_A} = \frac{15.75}{2} + \frac{60^2}{12(15.75)}$$

$$= 26.92 \text{ ft}$$

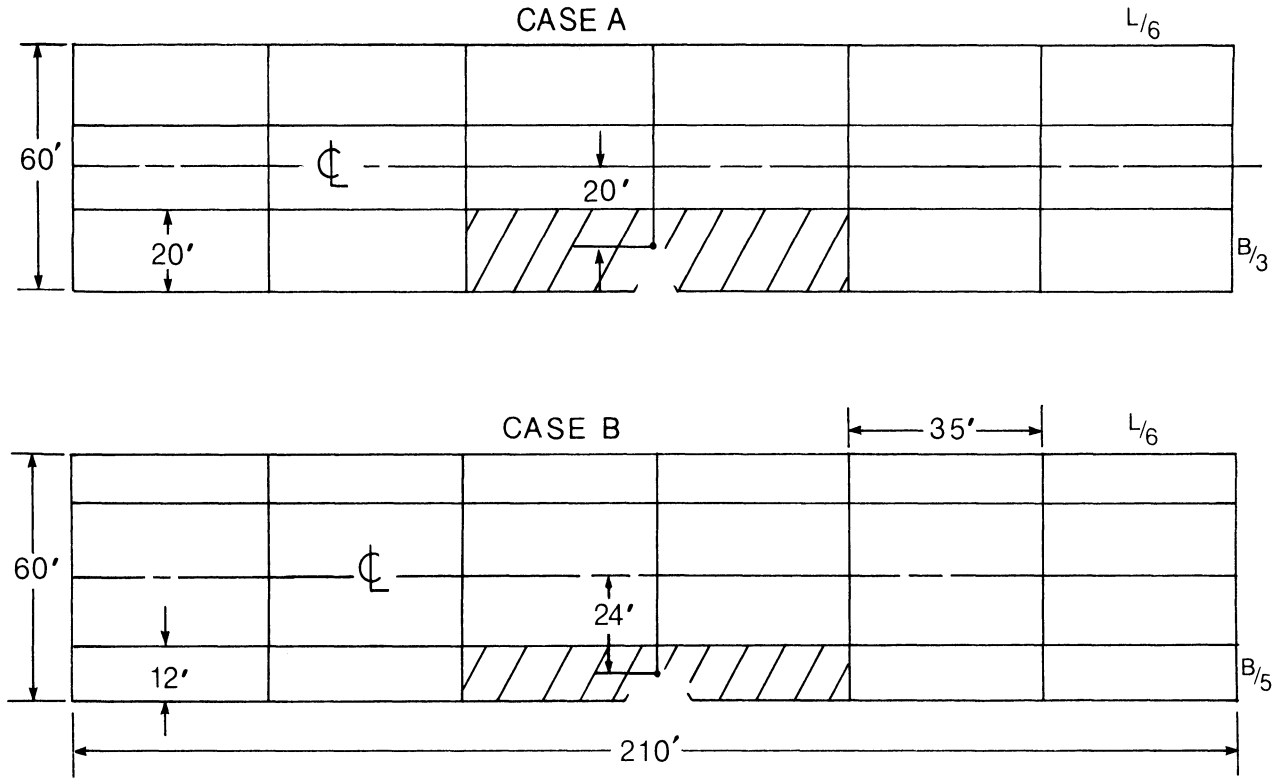


Figure 10-18. Rectangular barges viewed from above

$$\overline{KM}_B = \frac{T_B}{2} + \frac{B^2}{12T_B} = \frac{15}{2} + \frac{60^2}{12(15)}$$

$$= 27.5 \text{ ft}$$

$$\overline{GM}_A = \overline{KM}_A - \overline{KG}_A$$

$$= 26.92 - 11.1 = 15.8 \text{ ft}$$

$$\overline{GM}_B = 27.5 - 11.2 = 16.3 \text{ ft}$$

$$FSC_A = 2 \frac{i}{\nabla_A} = \frac{2\left(\frac{1}{12}\right)(35)20^3}{210(60)15.75}$$

$$= 0.24 \text{ ft}$$

$$FSC_B = 2 \frac{i}{\nabla_B} = \frac{2\left(\frac{1}{12}\right)(35)12^3}{210(60)15}$$

$$= 0.05 \text{ ft}$$

$$FCC_A = \frac{ay_A^2}{\nabla_A} = \frac{70(20)20^2}{210(60)15.75}$$

$$= 2.82 \text{ ft}$$

$$FCC_B = \frac{ay_B^2}{\nabla_B} = \frac{70(12)(24)^2}{210(60)15}$$

$$= 2.56 \text{ ft}$$

$$\overline{GM}_{\text{eff}} = \overline{GM} - FSC - FCC$$

$$\overline{GM}_{A\text{eff}} = 15.8 - 0.24 - 2.82$$

$$= 12.74 \text{ ft}$$

$$\overline{GM}_{B\text{eff}} = 16.3 - 0.05 - 2.56$$

$$= 13.69 \text{ ft}$$

From equation 10-13,

$$\tan \phi = \frac{wy}{\Delta \overline{GM}_{\text{eff}}}$$

$$\tan \phi_A = \frac{630(20)}{5,670(12.74)} = 0.174$$

$$\phi_A = [9.9^\circ]$$

$$\tan \phi_B = \frac{360(24)}{5,400(13.69)} = 0.117$$

$$\phi_B = [6.7^\circ]$$

Positioning the longitudinal bulkhead further from the centerline (case B) resulted in a smaller free-communication correction and a smaller free-surface correction. The reduction in the amount of flooding water

w reduces the heeling moment sufficiently to decrease the angle of list. It can be shown that the position of the longitudinal bulkhead in case A yields the maximum free-communication correction. Note that the list angle for case A is near the limit of applicability of equation 10-13 and represents only an approximate solution. To keep this example within the range of applicability, a large $\overline{GM}$ was required, which necessitated a relatively low value of $\overline{KG}$ and a modest draft. Increases in either draft or $\overline{KG}$ would require the use of the wall-sided-ship method of analysis (equation 8-9) to account for the shifting location of the metacenter M , or the use of cross curves of stability to obtain a corrected static-stability curve.

Problems

10-1. A ship enters overhaul with a weight of 7,500 tons, a $\overline{KG}$ of 24 feet, and a center of flotation 40 feet aft of midships. The following alterations are made:

Additions	$\overline{KG}$	Distance from centerline	Distance from midships
100 tons	10 feet	10	60 feet aft
Deletions			
300 tons	25 feet	15	30 feet aft

- How much ballast can be added without increasing displacement?
- At what $\overline{Kg}$ can the ballast be added without increasing $\overline{KG}$?
- How far off the centerline must the ballast be added to prevent a permanent list?
- Where must the ballast be added to prevent a permanent trim?

10-2. A ship inadvertently runs aground. Preliminary calculations yield the following:

Before grounding	At low tide
$T_m = 22$ feet	$T_m = 19$ feet
buoyant force = 4,200 tons	buoyant force = 2,940 tons
$\overline{KG} = 19$ feet	$\overline{KM} = 26.3$ feet

Is the ship initially stable at low tide?

10-3. A rectangular barge is used for towing a 12,000-ton offshore oil platform out to the drilling site. The barge is 630 feet long, 100 feet wide, and 30 feet high as illustrated in figure 10-a. When loaded with the oil rig, the draft is 9 feet and the $\overline{KG}$ is equal to 60 feet.

The barge is divided into ten watertight compartments as shown, with five on each side of the centerline. A collision with a submerged object on the way out to the site holes one of the compartments, causing a free-surface and free-communication condition. As a result, the mean

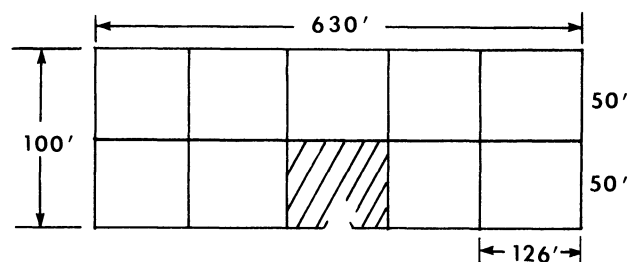


Figure 10-a

draft increases to 10 feet and the $\overline{KG}$ is lowered to 55 feet.

- Find the displacements before and after damage.
- Find the $\overline{GM}$ before and after damage.
- Find the angle of list.

10-4. A tramp steamer displacing 19,000 tons in ballast condition collides with another vessel, opening up a previously empty compartment to free communication with the sea. One thousand tons of seawater flood the 35-ft-wide by 50-in-long compartment with 20 feet of water. The center of gravity of the flooding water is 20 feet to starboard and 40 feet forward of amidships. (The LCF is 20 feet aft of amidships.) Assume the actual $\overline{KG}$ is lowered from 16 feet to 15.75 feet and $\overline{KM}$ remains at 19 feet.

- Determine the free-surface and free-communication corrections and the final effective $\overline{GM}$.
- The ship is 600 feet long on the waterline and the original mean draft was 21 feet. Find the final drafts fore and aft ($TPI = 100$ tons/inch and $MTI = 5,000$ ft-tons/inch).

10-5. A DD 692-class destroyer suffers extensive damage from an explosion and fire. The fire in the galley and messing spaces resulted from the explosion of a terrorist bomb hidden in locally purchased food supplies. The fire has burned out of control for some time and fire-fighting parties have used sufficient water to cause the vessel to trim 1 foot by the bow and to take on a list of 3° to port. From the increase in mean draft, the damage-control assistant estimates that the ship now displaces 100 tons more than when the fire fighting started. He also estimates that 30 tons of the water used to fight the fire has drained down into the crew's quarters, A-306-L, just below the food-service area where the explosion took place. The center of gravity of the 30 tons of water is 11 feet to port of the centerline.

Most of the rest of the water is located in the crew's messroom, A-205-L, which extends 30 feet from one side of the ship to the other and is 22 feet long, giving an extensive free-surface area.

Using the data listed in the table below, determine the free-surface correction and both the actual and effective $\overline{GM}$. Plot the general-stability diagram for the damaged condition and indicate the maximum righting arm and the range of stability to port.

	<i>Before damage</i>	<i>After damage</i>
<i>Mean draft</i>	14 feet 2 inches	14 feet 6 inches
<i>Draft forward</i>	14 feet 2 inches	15 feet 0 inches
<i>Draft aft</i>	14 feet 2 inches	14 feet 0 inches
<i>Displacement</i>	3,400 tons	3,500 tons
<i>List</i>	0°	3° to port
$\overline{KM}$	19.2 feet	19 feet
$\overline{KG}$	15.5 feet	

10–6. While patrolling in the Persian Gulf, a DD 692-class destroyer takes a guided-missile hit in the after engine room and an adjacent off-center space. The engine room takes on seawater rapidly and settles by the stern. The damage-control assistant estimates that the ship took on 390 tons of seawater whose center of gravity was 67.5 feet aft of amidships, 9 feet above the keel, and 2.2 feet to port. The area subject to free surface and free communication is 40 feet wide and 30 feet long (neglect permeability for this problem).

Before damage, the ship displaced 3,010 tons on an even keel with a mean draft of 13 feet. The $\overline{KG}$ before damage was 15.6 feet.

- Determine the mean draft after damage.
- Determine the free-surface and free-communication corrections.
- Determine the $\overline{GM}_{\text{eff}}$ after damage.

- Determine the angle of list.
- Determine the final drafts fore and aft after damage.

10–7. A DD 692-class destroyer is involved in a collision on the port side that holes the ship between frames 60 and 72 with 100 tons of seawater. The center of gravity of the flooded compartment is 11.5 feet above the keel, 12 feet to port of the centerline, and 80 feet forward of amidships. The flooded compartment is 11 feet wide and 22 feet long, has a free surface, and is in free communication with the sea.

Before the collision, the draft forward was 13 feet 9 inches and the draft aft was 13 feet 11 inches. The displacement was 3,300 long tons with a $\overline{KG}$ of 15.1 feet.

- Determine the final $\overline{GM}$ caused by the seawater flooding (without list).
- Determine the angle of list.
- Determine the change in trim and the final drafts fore and aft.

10–8. Write a computer program based on the analysis done in example 10–1 to illustrate the effect on $\overline{GM}_{\text{eff}}$ and the angle of list as a function of the number and width of the compartments with longitudinal bulkheads. Consider two cases: a) the damage and free communication are limited to two adjacent outer compartments, and b) the damage and free communication occur in two adjacent compartments longitudinally and in the next two inner compartments (this can be related to the side-protection discussion in section 10–2).

Ship Resistance and Powering

11-1 Introduction to Hydrodynamics

The term hydrodynamics was introduced by Daniel Bernoulli (1700–1783), the man who also discovered the fundamental theorem relating velocity, pressure, and elevation for the steady, inviscid (frictionless) flow of an incompressible fluid along a given streamline.

$$p + \frac{1}{2} \rho v^2 + \rho g z = C_1 \quad (11-1)$$

where p = fluid pressure
 ρ = fluid density (constant)
 v = fluid velocity
 z = elevation relative to some reference axis
 g = acceleration of gravity
 C_1 = Bernoulli constant for a particular streamline.

If the fluid flow is also *irrotational* (lacking *vorticity*), the Bernoulli constant is the same for all streamlines. The Bernoulli theorem is customarily used to explain how *lift* (a force perpendicular to the direction of motion) is generated by aircraft wings, hydrofoils, sails, propellers, and control surfaces. If one considers the irrotational flow around a wing-like object, as illustrated in figure 11-1, the Bernoulli theorem constant applies to all the streamlines outside the *boundary layer* (to be defined later), where the effects of fluid viscosity are concentrated. Since the fluid must accelerate to higher velocity along the longer path over the top of the wing, equation 11-1 indicates that the pressure must decrease where the velocity increases and vice versa. Thus the pressure on the bottom side of the wing is greater than on the top side, and lift is produced.

Other significant contributions to theoretical hydrodynamics were made during the eighteenth century by J. R. d'Alembert (1717–83), L. Euler (1701–83), J. L. Lagrange (1736–1813), and others whose work applied

to *frictionless fluids*. C. L. Navier (1785–1836) and G. G. Stokes (1819–1903) developed the equations of motion for the flow of viscous fluids, but little progress was made in the nineteenth century in relating this work to the practical aspects of hydraulics and ship resistance. Hydraulics engineers, concerned with pumping systems for fluid flow in pipes and conduits and with flow measurement, developed an empirical science based on experimental data that was unrelated to the work in theoretical hydrodynamics. Ship designers and builders of sailing vessels, on the other hand, had only rules of thumb based on the midship-section area to size the sails for a given vessel. The hull form and sail plan for the clipper ships, for example, evolved from experience, not from theory. Shipowners had to accept whatever speed could be obtained from the number and size of sails the ship could carry in given weather conditions.

11-2 Steam Propulsion

It was not until the advent of steam power and the construction of large iron steamships in the mid 1800s that it became clear to shipowners and shipbuilders that a more rigorous approach to the design of ship hulls and propulsion systems was needed. There was controversy in England over the amount of steam power required to drive a vessel. Many were convinced that resistance was directly proportional to displacement, and therefore it was not possible to build a steam-powered ship that could cross the Atlantic without refueling. Others, led by the famous English railway and bridge engineer Isambard K. Brunel (1806–59), were convinced of the economic advantage of large vessels. In 1836, Brunel stated: “It is well known that the proportional consumption of fuel decreases as the dimensions and power of engines are increased and consequently a large engine can be much more economical than a small one. The resistance of vessels on the water

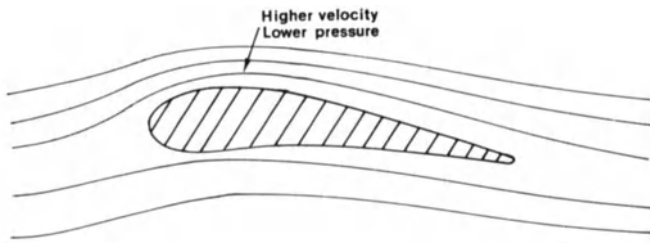


Figure 11-1. Stream lines around a wing.

does not increase in direct proportion to the tonnage. The tonnage increases with the cubes of their dimensions while the resistance increases at about their squares, so that a vessel of double the tonnage of another capable of containing an engine of twice the power does not really meet with twice the resistance. Speed therefore would be greater with the larger vessel or the proportion of power in the engine and consumption of fuel may be reduced" (Corlett 1974, p. 10).

The statement, of course, applies only to large vessels traveling at slow speeds for their size, but it explains the reason for the economic advantage of the large supertankers and bulk carriers of today. Based on this principle, Brunel proposed and built the 2,300-ton (displacement) paddle-powered steamship *Great Western*

in 1838, the first to be designed specifically to steam across the Atlantic. The *Great Western* could travel at just over 9 knots on her nominal 400-hp engines alone, and made many voyages across the Atlantic with coal to spare. The success of the *Great Western* led Brunel to design the *Great Britain*, the first large iron steamship, to cross the Atlantic. The *Great Britain* was 322 feet long, displaced 3,400 tons, and was the first large screw-propelled steamer. Her propulsion plant was sized by scaling up the *Great Western* performance using rules of thumb and by propeller tests using data from the screw-driven *Archimedes*. Brunel predicted that the ship would travel at a speed of 12 knots for the trials and at one knot for each $1\frac{1}{2}$ revolutions per minute of the engine. The *Great Britain* actually made about $11\frac{3}{4}$ knots at 18 revolutions per minute, so Brunel's predictions were surprisingly accurate (Corlett 1974, p. 105).

In order to consider even longer voyages to Australia and the Far East and back, Brunel next proposed the *Great Eastern*, a 693-ft-long monster with a full-load displacement of 27,000 tons, including large coal bunkers. He assumed that the ship would require a horsepower of $1\frac{1}{4}$ times the midships-section area, but only predicted that the ship would have "a very high speed." She was powered by paddle wheels driven by 1,000-hp engines, a 24-ft-diameter screw propeller driven by a 1,600-hp engine, and 6,500 square yards of sail. The *Great Eastern* took six years to build and three months to launch, and became a financial disaster before ever being put into service.

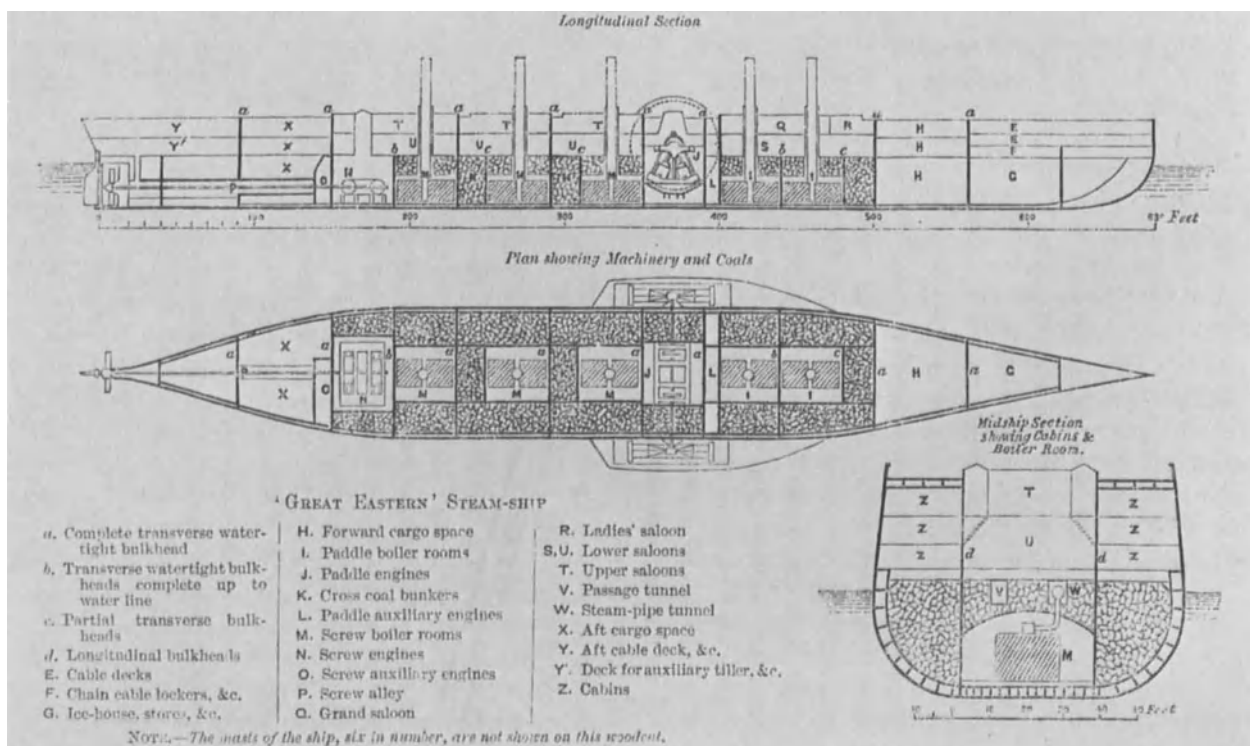


Figure 11-2. The *Great Eastern*. (From Brunel 1972.)

Her design included innovations such as a double skin and ten watertight transverse bulkheads, as shown in figure 11-2. The *Great Eastern* was under powered by today's standards, but did manage to make 14 knots on a calm day during her maiden voyage to America in 1860. Unfortunately, because of mismanagement, bad luck, and inefficient utilization of her capabilities, she continued to be an economic disaster for her backers. Her size was finally justified when she was employed to lay the first successful transatlantic telegraph cable in 1866 and several other transoceanic cables, a task for which her huge cargo capacity was ideally suited. (She was once loaded to 34,000 tons.) That task completed, she was retired for exhibition as a curiosity and a showboat. She was scrapped in 1889 but remained the largest ship in the world for forty years (Brunel 1971).

The indirect contribution of the *Great Eastern* to the history of experimental naval architecture lay in her association with William Froude (1810-79). Froude had worked with Brunel off and on since 1837 and shared his interest in discovering the secrets of predicting ship performance. His primary interest until 1865, however, was the subject of ship rolling. He had been aboard the *Great Eastern* during her trials and had suggested the addition of bilge keels to reduce rolling, a suggestion that was adopted.

In 1868 W. Froude participated in a study entitled "The Stability, Propulsion, and Seagoing Qualities of Ships." He was the only member of the committee who did not dismiss the idea of ship-model testing. The rest of the committee was committed to full-scale trials, including towing tests, which caused Froude to state: "I contend that unless the reliability of small-scale experiments is emphatically disproved, it is useless to spend vast sums of money upon full-scale trials, which, after all, may be misdirected, unless the ground is thoroughly cleared beforehand by an exhaustive investigation on a small scale" (Froude 1955, p 133).

11-3 Froude's Experiments and the Law of Comparison

In the fall of 1867, Froude experimented with pairs of 3-, 6-, and 12-ft-long models towed by a steam launch at the mouth of the River Dart. Froude concluded that the wave patterns he observed were significantly related to ship resistance, and that there was no proportional or mathematically describable relationship of the observed resistance as a function of size or cross section alone, as was the case for the rule-of-thumb Admiralty constants used by Brunel and other ship designers for ship-powering calculations.

Part of the ship-powering problem was the apparent wave disturbance made by the model or the ship moving through water. These self-created waves obviously represented an energy loss. A rigid object moving through

water will move that water aside, a displacement which creates a pattern of forces not existing when the object is at rest. Froude also observed that the pattern of the waves was a function of the model's speed—small, closely spaced wavelets developed when the model was moving slowly, and widening, larger waves developed as speed increased. Most importantly, these patterns were similar, regardless of model size. It became apparent that there must be a relationship between wave-making resistance and velocity, and this might very well be associated with the velocity of a free wave, where the speed of the wave (V) is proportional to the square root of its wave length ($\sqrt{L}$).

In fact, Froude had observed geometrically similar wave patterns at what he called *corresponding speeds*, that is, when both model and ship travel at the same speed-length ratio, or $V/\sqrt{L}$. On the basis of these observations, he postulated the original form of his law of comparison in 1868. "We are thus brought to the scale of comparison which was just now enunciated, that the entire resistances of a ship, and similar model are as the cubes of their respective dimensions if their velocities are as the square roots of their dimensions" (Froude 1955, p. 131).

The original law of comparison attempted to scale the total resistance, but this was later changed to exclude frictional resistance. Froude realized that a resistance component that was just as important as wave-making resistance was the frictional resistance of the molecules of water being dragged along next to the hull. If the amount of this friction could be separately evaluated, and if there were no more major components of the total resisting forces, then it might be possible to relate the model- and parent-ship characteristics. Consequently he made a proposal to E. J. Reed of the British Admiralty to construct a towing tank at Torquay for £2,000. Froude's memorandum included a methodical set of proposed experiments designed to establish a more rigorous method of predicting ship-powering requirements. Since model tests measure total resistance, he first proposed to carry out a set of tests on skin-friction resistance, followed by a series of tests on standard hull forms in use by the British navy at that time. The Admiralty funded his proposal with the stipulation that he also carry out a set of experiments on ship rolling. Froude built a towing tank, 278 feet long and 36 feet wide, with a small carriage placed on an overhead rail and powered by a steam winch. He was now able to control his model's speed more precisely and measure total drag or resistance on a beam balance scale. Actually, he had built the prototype of all modern towing tanks. This first essential, but rather primitively instrumented tool, soon began to open many of the previously locked-up secrets of experimental hydrodynamics.

Froude conducted tests on flat-surfaced, comparatively thin planks up to 50 feet long, towed on edge just below the water's surface. In such tests, negligible waves formed. Using planks with varieties of surface finishes, from smooth varnish to sand, cotton, paraffin, etc., he soon discovered that there were several variables affecting friction. Important among them were velocity, area of surface, smoothness of surface, and viscosity and density of water. Froude plotted the results on curves of frictional resistance versus speed for the various lengths and surface finishes. He and his son, R. E. Froude, who took over the tank following his father's death in 1879, developed a better series of curves, based on skin-friction resistance per unit area for various surfaces. They also fitted the following expression to the data, so the results could be extrapolated to the wetted-surface area of a full-scale ship:

$$R_f = fSV^n \quad (11-2)$$

where R_f = frictional resistance of the fluid
 S = wetted surface
 V = velocity
 f and n depend on the length and nature of the surface.

Note that n has a constant value of 2.0 for roughened planks, and a value of about 1.825 for long, smooth, varnished planks. Also, f decreased in value as the length of a smooth plank increased, and increased for rough surfaces.

The Admiralty also requested that William Froude carry out a set of full-scale trials, by towing 172-ft-long HMS *Greyhound* at full scale and comparing the component resistances with those obtained from model tests of the same hull. As it turned out, the actual measured ship resistance was always higher than that predicted from the model tests, but the shape of the curves of scaled resistance plotted against scaled speed was essentially the same.

Froude attributed the additional resistance of the ship to differences in surface roughness (which is still true for present towing-tank methods) and added the hypothesis of *equivalent flat-plate resistance*. "For this calculation, the immersed skin was carefully measured, and the resistance due to it determined upon the hypothesis that it is equivalent to that of a rectangular surface of equal area, and of length (in the line of the motion) equal to that of the model moving at the same speed" (Froude 1955, p. 248). In other words, the frictional resistance of either the model or the prototype is equivalent to a flat plate having the same length and wetted surface traveling at the same speed.

In 1877 Froude presented a paper to the Institution of Naval Architects, in which he defined the *residuary resistance* to include both wave-making and eddy-making

resistance. He had noted that the eddy-making resistance constituted about 8 percent of the skin-friction resistance of some of the hulls he had tested. However, if he lengthened the *parallel middle body*, the eddy resistance did not change in proportion to the skin friction, so he combined eddy-making with the wave-making resistance to form the residuary resistance, which could be scaled up by the redefined law of comparison. Froude also initiated the search for the optimum hull form for a given speed that would minimize the combination of skin-friction, wave-making, and eddy-making resistance. The fact that Froude was able to establish the basis for modern-day ship-model testing without the benefit of either Osborne Reynolds's discoveries concerning the nature of laminar and turbulent flow (published in 1883) or the work of L. Prandtl in the early 1900s on the concept of the boundary layer, makes his work one of the breakthroughs in the history of engineering design.

Froude's method for predicting full-scale ship resistance can be summarized in the following steps:

1. Construct a geometrically similar model and measure its resistance-versus-speed characteristics at the appropriate corresponding speeds (speed-length ratio $V/\sqrt{L}$).
2. Estimate the equivalent flat-plate resistance of the model at each speed, based on the plots of plank resistance or values of f and n . (This estimate is subject to several sources of error to be discussed later.)
3. Estimate the residuary resistance of the model by subtracting the equivalent flat-plate resistance from the total resistance at each speed.
4. Estimate the residuary resistance of the full-scale ship by multiplying each value of the model residuary resistance by the ratio of the ship-to-model displacements. (This is another improvement of the original law of comparison to account for the density difference between the seawater and that in the towing tank [see equation 11-11].)
5. Estimate the equivalent flat-plate resistance for the ship at each corresponding speed, based on extrapolated plank-resistance data. (Judicious use of allowances based on experience could improve the estimate.)
6. Estimate the total ship resistance by adding the equivalent flat-plate resistance to the residuary resistance at each speed-length ratio.

William Froude's method of extrapolating scale-model test data to predict full-scale ship resistance forms the basis for current practice in towing tanks around the world. The use of dimensionless coefficients derived from dimensional analysis of the problem and a reformulation of the method of estimating equivalent flat-plate resistance have improved ship-performance prediction to the point that it is a well-respected design tool.

11-4 Dimensional Analysis and Dynamic Similitude

Dimensional analysis is a useful tool for correlating experimental data when an exact relationship between the variables is not known. It involves the use of *dimensionless numbers* to reduce the number of variables that must be investigated to determine the functional relationships in experimental data. Dimensionless numbers were first used in fluid mechanics by Osborne Reynolds in 1884, when he developed what is now called the Reynolds number to characterize the difference between laminar and turbulent flow. J. B. Fourier laid the foundations of dimensional analysis in 1888, but E. Buckingham popularized what is called the π theorem in 1915.

π THEOREM

Suppose a physical problem involves n variables that have a total of m different fundamental dimensions, such as mass (M), length (L), and time (T). Then

$$F(\pi_1, \pi_2, \pi_3, \dots, \pi_{n-m}) = 0$$

or

$$\pi_1 = G(\pi_2, \pi_3, \dots, \pi_{n-m})$$

where $\pi_2, \pi_3, \dots, \pi_{n-m}$ are independent dimensionless ratios. The π theorem merely states that there may exist a functional relationship between the π terms. Experimental data must be analyzed to determine the relationship.

Several procedures are available for establishing the π parameters. The method demonstrated below involves the use of *repeating variables* equal to the number of basic dimensions in the problem.

Example 11-1: Ship resistance for surface ships is considered to be a function of the following variables:

$$R_T = F(\rho, v, L, g, \mu) \quad (11-3)$$

where R_T = total ship resistance in lb or ML/T^2

ρ = water density in slugs/ft³ or M/L^3

v = ship velocity in ft/sec or L/T

L = a characteristic length in ft or L

g = acceleration of gravity in ft/sec² or L/T^2

μ = absolute viscosity in slugs/ft-sec or M/LT

Note that there are only three basic dimensions in this problem, since forces can be made equivalent to ML/T^2 from Newton's second law,

$$F = ma$$

Thus, there will be a total of three dimensionless parameters as follows:

$$6 - 3 = 3$$

The repeating variables must include all the basic dimensions, and no two of them should have the same net dimensions. It has been found useful in fluid mechanics to select ρ , v , and L as the repeating variables.

The parameters are determined by setting up dimensional equations, based on the repeating variables plus one of the remaining variables in turn.

$$\pi_1 = \rho^a v^b L^c R_T^1$$

$$M^0 L^0 T^0 = \left(\frac{M}{L^3}\right)^a \left(\frac{L}{T}\right)^b L^c \frac{ML}{T^2}$$

Equating exponents for each dimension,

$$M: 0 = a + 1$$

$$L: 0 = -3a + b + c + 1$$

$$T: 0 = -b - 2$$

Thus,

$$a = -1$$

$$b = -2$$

$$c = -2$$

Therefore,

$$\pi_1 = \frac{R_T}{\rho v^2 L^2}$$

$$\pi_2 = \rho^a v^b L^c g^1$$

$$M^0 L^0 T^0 = \left(\frac{M}{L^3}\right)^a \left(\frac{L}{T}\right)^b L^c \frac{L}{T^2}$$

$$M: 0 = a$$

$$L: 0 = -3a + b + c + 1$$

$$T: 0 = -b - 2$$

Therefore,

$$a = 0$$

$$b = -2$$

$$c = 1$$

$$\pi_2 = \frac{gL}{v^2}$$

and

$$\pi_3 = \rho^a v^b L^c \mu^1$$

$$M^0 L^0 T^0 = \left(\frac{M}{L^3}\right)^a \left(\frac{L}{T}\right)^b L^c \frac{M}{LT}$$

$$M: 0 = a + 1$$

$$L: 0 = -3a + b + c - 1$$

$$T: 0 = -b - 1$$

Therefore,

$$\begin{aligned} a &= -1 \\ b &= -1 \\ c &= -1 \\ \pi_3 &= \frac{\mu}{\rho v L} \end{aligned}$$

Since π_1 equals $F(\pi_2, \pi_3)$,

$$\frac{R_T}{\rho v^2 L^2} = F\left(\frac{gL}{v^2}, \frac{\mu}{\rho v L}\right) \quad (11-4)$$

The addition of the constant $\frac{1}{2}$ in front of the density is seen to convert π_1 into the basic equation for fluid dynamic drag; that is,

$$2\pi_1 = C_D = \frac{\text{drag}}{\frac{1}{2}\rho v^2 A} \quad (11-5)$$

or

$$C_T = \frac{R_T}{\frac{1}{2}\rho v^2 S} \quad (11-6)$$

In this formulation, L^2 has been replaced by the wetted-surface area S , or the wing (or frontal) area A . C_T is called the *total resistance coefficient* and C_D is called the *drag coefficient*. (The lift coefficient C_L is found by substituting the lift force for the drag force in equation 11-5.)

If π_2 is inverted and its square root taken, one obtains what is called the Froude number,

$$\text{Froude number} = F_n = \frac{v}{\sqrt{gL}} \quad (11-7)$$

in honor of William Froude, who first put forth the similar concept of the corresponding speed $V/\sqrt{L}$, which is not a dimensionless number since g was omitted.

The inverse of the Reynolds number discussed in section 11-6 is seen to be π_3 .

$$\text{Reynolds number} = R_n = \frac{\rho v L}{\mu} \quad (11-8)$$

Thus, equation 11-4 can be rewritten as

$$C_T = F(F_n, R_n) \quad (11-9)$$

and the problem of experimental-data correlation has been reduced from one dependent variable and five independent variables (equation 11-3) to a single dimensionless coefficient that is a function of two dimensionless parameters. The functional relationship is not a simple one, however, as Froude had already discovered before the technique of dimensional analysis was

developed. Note that the factor f in Froude's equation 11-2 for flat-plate resistance was not dimensionless and must have included the effect of density.

DYNAMIC SIMILITUDE

In order for model tests to precisely predict the performance of full-scale vehicles such as ships and aircraft, the following conditions of similitude have to be satisfied:

1. *Geometric similitude*, which requires that all length ratios be the same. This is the concept behind a scale model.

2. *Kinematic similitude*, which requires that the flow fields around the model and prototype have scaled magnitudes and identical directions at the corresponding locations.

3. *Dynamic similitude*, which requires that the forces associated with the fluid motion around both the model and prototype have scaled magnitudes and identical directions at the corresponding locations.

A common method for evaluating whether or not dynamic similitude is present is to determine if all the dimensionless parameters identified for a particular problem have the same values for both model and full-scale prototypes. It is seen from equation 11-9 that complete dynamic similarity for model tests of geometrically similar surface ships requires both to have the same Froude number and Reynolds number. This is impossible, since the Reynolds number has a velocity-length product and the Froude number varies with velocity divided by the square root of length. This is the reason that little confidence was given to ship-model tests before Froude's methods were introduced. His hypothesis of separating total resistance into equivalent flat-plate resistance and residual resistance implies the following functional dependence:

$$C_T = C_F(R_n) + C_R(F_n) \quad (11-10)$$

where C_F = frictional-resistance coefficient assumed to be a function of Reynolds number only

C_R = residuary (residual) resistance coefficient assumed to be a function of Froude number only.

It also sets up model tests based on *incomplete dynamic similitude*, in which the tests are run on the basis of equal Froude numbers, and in which the effects of the Reynolds number differences are handled in a manner based on Froude's hypothesis concerning equivalent flat-plate resistance. (Model tests based on equal Reynolds numbers would require that the model velocity increase as the scale length decreases.)

The principal difficulty in this method is that the residuary resistance includes the eddy-making resis-

tance, which is generally thought to be a function of both Reynolds and Froude numbers. This will be discussed in more detail in section 11-7.

Froude's law of comparison for corresponding speeds (equal Froude numbers) can now be justified as follows, using subscript M for the model and subscript S for the ship:

$$\frac{v_M}{\sqrt{gL_M}} = \frac{v_S}{\sqrt{gL_S}}$$

or

$$\frac{v_M^2}{v_S^2} = \frac{L_M}{L_S} = \frac{1}{\lambda}$$

where $\lambda = \frac{L_S}{L_M}$ = scale ratio.

Modifying equation 11-6 to that for residual resistance,

$$\frac{R_{RM}}{R_{RS}} = \frac{C_{RM}\rho_M v_M^2 S_M}{C_{RS}\rho_S v_S^2 S_S}$$

but

$$\frac{S_M}{S_S} = \left(\frac{L_M}{L_S}\right)^2 = \frac{1}{\lambda^2}$$

$$\frac{R_{RM}}{R_{RS}} = \frac{C_{RM}}{C_{RS}} \cdot \frac{\rho_M}{\rho_S} \cdot \frac{L_M}{L_S} \left(\frac{L_M}{L_S}\right)^2$$

$$\frac{R_{RM}}{R_{RS}} = \frac{C_{RM}}{C_{RS}} \cdot \frac{\rho_M}{\rho_S} \left(\frac{L_M}{L_S}\right)^3$$

but

$$\left(\frac{L_M}{L_S}\right)^3 = \frac{v_M}{v_S} = \frac{1}{\lambda}$$

and

$$C_{RM} = C_{RS} \text{ for equal } F_n$$

$$\frac{R_{RM}}{R_{RS}} = \frac{\rho_M v_M}{\rho_S v_S} = \frac{\Delta_M}{\Delta_S}$$

or, as Admiral David W. Taylor, who built the first model basin in the U.S., preferred to present experimental data,

$$\frac{R_{RM}}{\Delta_M} = \frac{R_{RS}}{\Delta_S} \quad (11-11)$$

An example of this form of data presentation for both wave-making and frictional resistance is shown in figure 11-3.

11-5 Wave-Making Resistance

Were it not for wave-making resistance, the speeds of most ships would not be as embarrassingly slow as they are. They have improved over the centuries only by a small amount; sailing vessels can go no faster today than was possible a century ago; steam, diesel, or nuclear-powered surface vessels are held to speeds that were possible at the beginning of this century.

A ship moving through the otherwise undisturbed water's surface sets up a very characteristic pattern of waves. In this pattern, there are essentially two primary points of origin of waves, which are, logically enough, at the bow and at the stern (figure 11-4).

The bow wave train is more significant, because the waves generated here persist along the ship's hull, affecting the pressure distribution in the water where the ship is acting, and because the bow wave train is generally the larger and more predominant one (figure 11-5). In observing the characteristic pattern, a series of diagonal or oblique crests is first apparent. These are simply the diverging crests moving outwardly from the point of disturbance. (See Froude's sketch of a characteristic bow wave train in figure 11-6.) The system's basic wave train, of which the diverging crests are only a part, advances with the ship, and the line of crests and troughs is perpendicular to its course vector (Lord Kelvin first pointed out in 1887 that the envelope of the divergent wave crests makes an angle of $19^\circ 28'$ for a thin disturbance traveling in a straight line, regardless of the speed). It is these transverse waves, advancing with the ship and seemingly locked to it, that must be noted in relation to the ship's speed limitations and resistance characteristics. The length of these waves (the distance between crests in the direction of their movement) is, of course, a function of their velocity (phase velocity), which is also ship's velocity. Their amplitude, on the other hand, is independent of both velocity and wave length and is a function of the energy expended in generating them. This is apparent in observing the profile of these waves along or near the side of a moving ship (figure 1-11).

The energy associated with the transverse wave system travels at the "group velocity" of the waves, which equals one-half of the phase velocity in deep water. The propulsion system of the ship must therefore put additional energy into the wave system to replace that which "falls behind." A nominal relationship between ship speed and the length of the corresponding transverse wave may be found by equating the ship velocity with the *celerity* (phase velocity) of a small-amplitude gravity wave in deep water.

$$v_{\text{ship}} = c_{\text{wave}} = \sqrt{\frac{gL_w}{2\pi}} = 2.26\sqrt{L_w} \quad (11-12)$$

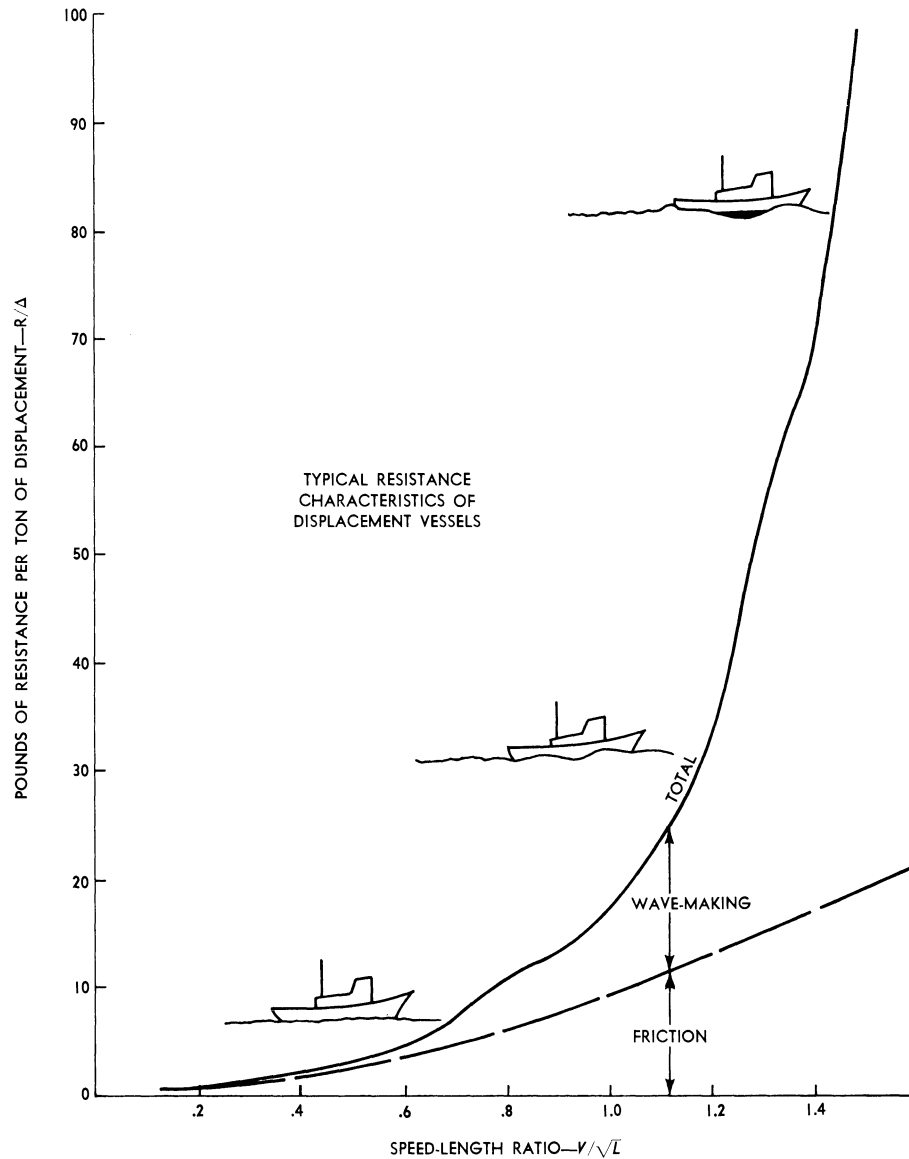


Figure 11-3. Resistance versus $V/\sqrt{L}$ in a characteristic speed-power curve for a displacement ship (note the ship-wave profiles)

where c_{wave} = celerity or phase velocity of the wave in ft/sec

L_w = length of the transverse wave in feet.

Equation 11-12 can be converted to speeds in knots by dividing by 1.688 fps/kt.

$$V_s(\text{knots}) = \frac{2.26}{1.688} \sqrt{L_w} = 1.34 \sqrt{L_w} \quad (11-13)$$

or

$$L_w = 0.56 V_s^2 \quad (11-14)$$

William Froude first pointed out the practical limiting speed for surface-displacement ships when he observed that “the speed at which wave resistance is accumulating most rapidly, is the speed of an ocean wave the length of which, from crest to crest, is about that

of the ship from end to end” (Froude 1955, p. 280). This condition is found by substituting the length of the ship for the length of the wave in equation 11-13, giving a relationship commonly referred to as the *hull speed*, or critical speed-length ratio.

$$\frac{V_s}{\sqrt{L_s}} = 1.3 \quad (11-15)$$

The equivalent Froude number for the critical speed-length ratio is found from equation 11-12.

$$\begin{aligned} F_n(\text{critical}) &= \frac{v_s}{\sqrt{gL_s}} = \frac{v_s}{\sqrt{gL_w}} \\ &= \frac{1}{\sqrt{2\pi}} = 0.40 \end{aligned} \quad (11-16)$$

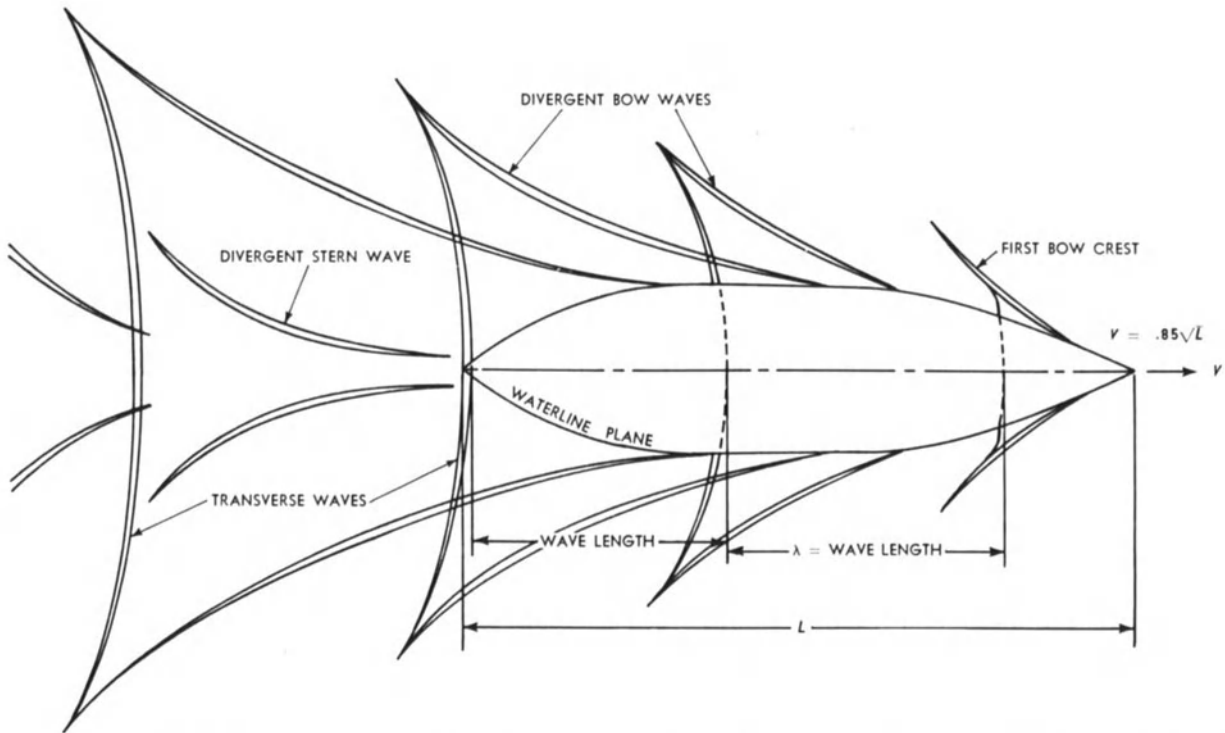


Figure 11-4. Characteristic wave pattern (plan view) as generated by a displacement ship when sub-hull speed V equals $0.85 \sqrt{L}$

When a surface ship attempts to exceed this speed, it finds itself literally climbing a hill of water. In exceptional cases of slim, highly powered ships such as destroyers, it is possible to exceed this speed, but it is seldom profitable.

Hull speed is somewhat analogous to the “sound barrier” for aircraft, which is not really a barrier but a practical limit for economic transportation. Near and above the speed of sound, an aircraft encounters additional wave drag caused by the formation of shock



Figure 11-5. Bow waves at point of origin, generated by the full bow of a Japanese 120,000-ton supertanker. (Courtesy of Hitachi Zosen.)

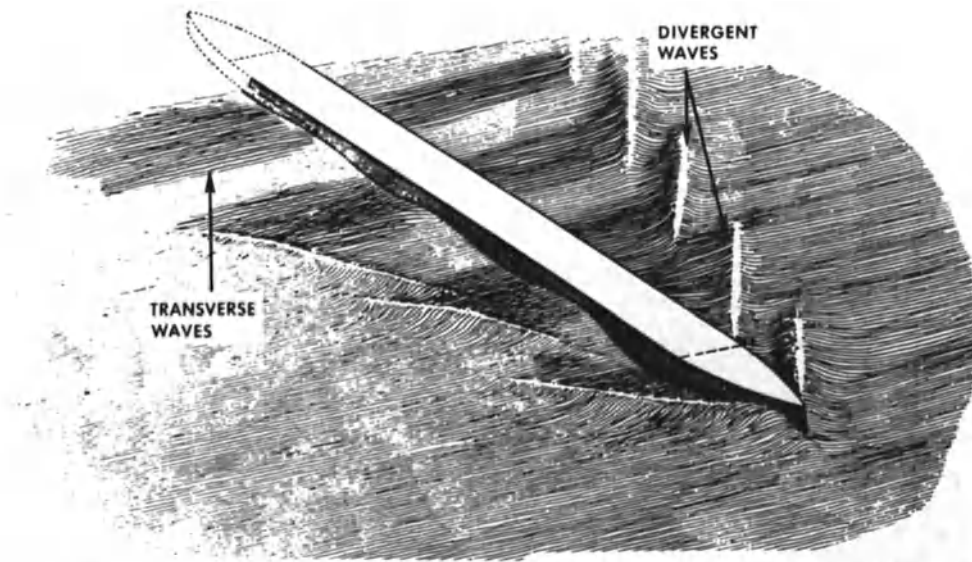


Figure 11-6. Froude's 1877 sketch of a characteristic wave train for ships. (From Froude 1955.)

waves, a phenomenon not unlike the divergent wave generated by a high-speed ship. A military mission can justify the installation of enough power to exceed these limiting speeds in the case of destroyers and jet fighters; however, the destroyer hull form and the shape of the jet fighter are both designed to minimize wave drag at high speeds.

A tugboat, which often moves at or near her hull speed when proceeding alone, generates a deep-troughed wave along her side. This reflects her full, deep hull and the amount of water that she must displace in relation to her length. All full, deep ships, such as tankers, tugs, trawlers, sailboats, submarines on the surface, and bulk-cargo carriers, generate wave patterns of pro-

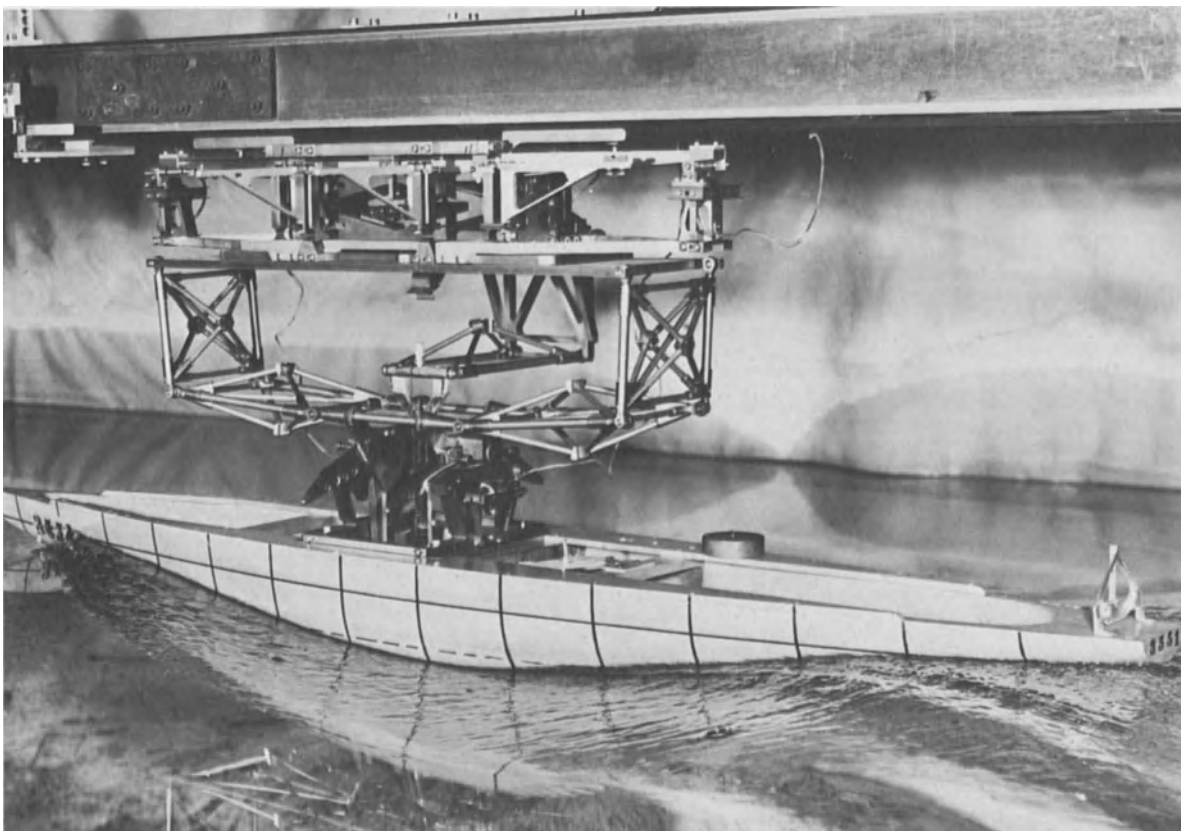


Figure 11-7. Full-hull model (12-meter sailboat, upright test) showing wave development just below hull speed $V/\sqrt{L} \approx 1.3$. (Courtesy of Stevens Institute of Technology.)

nounced, deep amplitude waves if moving in the higher speed-length ratios (figure 11-7). On the other hand, the shallower, narrower hulls with slim, sharp, forward waterlines generate comparatively shallow transverse wave trains. It is particularly difficult at times to even discern those of a naval destroyer-type vessel, and some craft move so easily and have so little displacement that overriding the quasi-limiting speed (a $V/\sqrt{L}$ of 1.3) is no problem. When such craft are designed with flat surfaces and straight buttocks, and their sections resolve the pressure vectors down and away laterally from the hull, they are able to rise dynamically and plane; they have greatly reduced wave-making resistance. Such craft are of limited commercial value, because they are unable to haul significant payloads or to negotiate sea conditions less than ideal.

In the search for higher speeds, some watercraft have been designed to remove themselves from the surface and its limitations. Hydrofoil vessels lift off the surface, planing craft skim on it, surface-effect craft are supported on a cushion of air above it, and new high-speed submersibles operate continuously below it.

These vessels also have limited payloads relative to displacement ships. The fact remains that surface transportation by displacement hulls is still the most practical and economical method for world commerce, and its disadvantages must be dealt with by ingenious ship designers.

BULBOUS BOWS

The search for means to reduce wave-making resistance has turned in many directions. Some solutions are fanciful and impractical, others are of far-reaching importance. Perhaps the most significant of these is the result of the development of a characteristic appendage or shape that extends back to warships of antiquity. As early as the classic period of the ancient Greeks and the seasoned Phoenicians, perhaps as early as 700 or 800 B.C., there is evidence of a protuberance at the lower forefoot of a ship, extending substantially ahead of the main structure of the hull. Because of its metal-clad sharpness this distinctive feature was obviously a ram. The ram was so successfully used that it was carried on war galleys for centuries and was used by Greek, Roman, Genoese, and Venetian galleys. It was less successful when war vessels propelled by oars were largely abandoned in favor of the vertically structured fortresses that carried hundreds of cannons and were propelled by sails.

With the advent of mechanical propulsion in the nineteenth century, the ram bow soon reappeared and, with other developments in the warship-design revolution, became a distinctive feature of the emerging "dreadnought" type of ship at the advent of the twentieth century. It was on hulls of this type that D. W.

Taylor had experimented in the old navy model basin in Washington, and in which he discerned a reduction of wave resistance and a hydrodynamic superiority in higher speed ranges. Taylor was convinced that a bulbous nose located deep at the forefoot and of rounded, faired-in form would produce less wave drag because of a newly created pressure pattern in the vicinity of the bow wave. Such a bow was designed and built specifically with this feature on the USS *Delaware* in 1907. The performance of the ship fully justified this old shape, and from that date large American-built ships of reasonably high power were built with bulbous bows. Many large foreign-built ships likewise adopted this distinctive form, but it is significant that two great ocean queens of the past forty years, the *Queen Elizabeth* and the *Queen Mary*, had the bulb omitted in their design (as did many other large British ships).

The bulbous bow, as conceived by Taylor and applied to large ships, was subsequently understood and designed to augment speed in the higher values of $V/\sqrt{L}$. The range was determined to be from 0.9 to 1.9 by various contemporary studies. However, on some ship forms, it was noted that the advantages became evident at lower speeds. Most merchant ships and cargo carriers are economically powered to cruise at speeds no higher than a $V/\sqrt{L}$ of 0.8. Consequently, with the prevailing theories surrounding the bulbous bow as well as the added cost in their construction, many ships did not have them and so were excluded from whatever advantages might have ensued.

In 1962 T. Inui advanced some remarkable theories and test results of his studies of wave-making resistance. These theories were the result of a number of years of research in the model basin at the University of Tokyo, where Inui had developed new and ingenious techniques of analysis. His studies led him through various configurations of bulb forms not only at the bow but at the stern, where the extensions concentrated on wave cancellation and speed augmentation in the moderate to low Froude number values (Inui 1962).

Inui's experiments, with his associated techniques and theoretical support, have shown substantial and in many cases remarkable reductions in resistance when his bulbous bows have been used. In fact, his work has been so enthusiastically accepted that not only are these bulb shapes incorporated in most new cargo and tanker constructions, but also many older ships have undergone conversion to take advantage of the shape.

It is interesting to compare the Inui Bow with the earlier bulbous bow introduced by Taylor. The Taylor Bow was conceived and is still used to augment speed characteristics in speed-length ratios of 1.0 and higher. The Inui Bow was developed for ships of greater range in speed, and its advantageous effects in some instances overlap the ranges of the former. In the lower speed

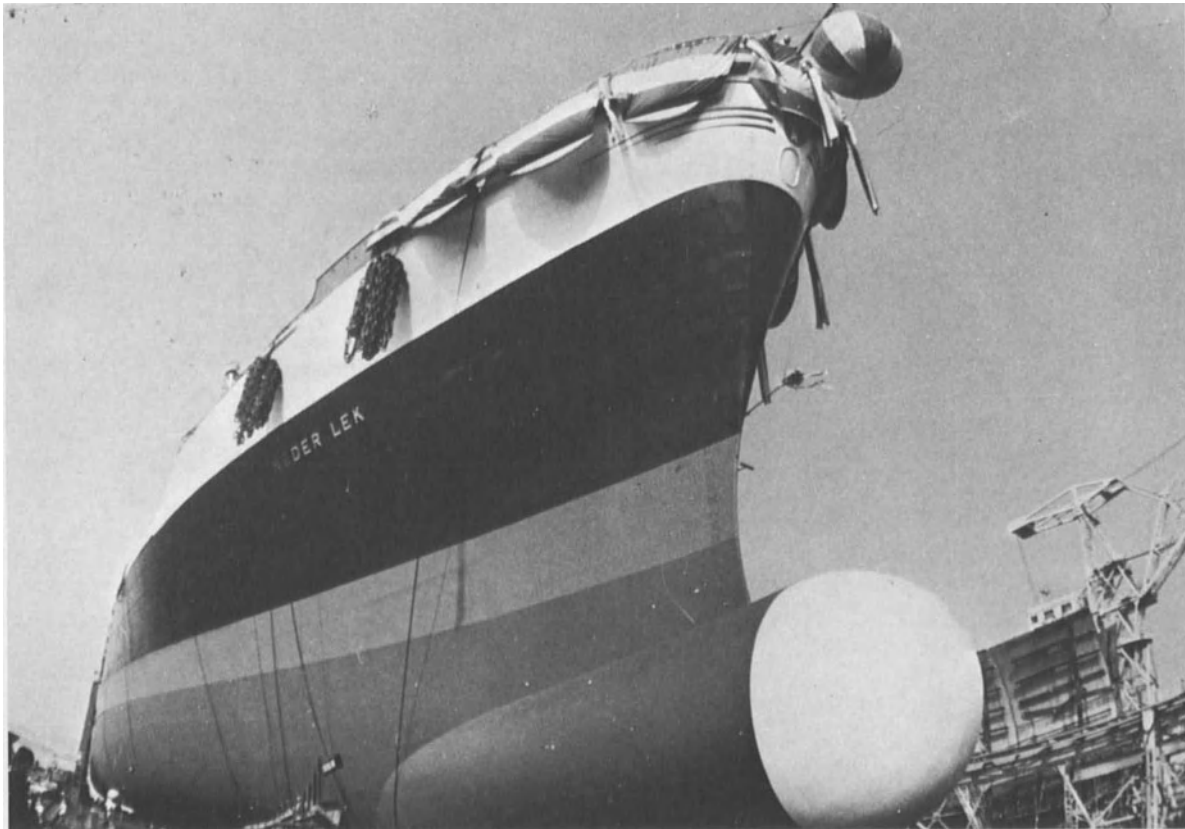


Figure 11-8. Bulbous bow characteristic of the *Inui* type. This projecting bulb is on a relatively high-speed cargo ship with a sharp waterline entrance. Higher block-coefficient tankers use a less projecting bulb, which is faired into the hull.

categories, a curious phenomenon exists with the new-type bulbs (figure 11-9). While it has been emphasized that these bulbous bows are forms applied to the forward extremities of the ship for the prime purpose of altering and, in a sense, neutralizing the pressure field there to reduce wave-making resistance, it would then seem natural that the advantages would diminish with lower speeds as the percentage of wave-making resistance diminishes. At speed-length ratios of 0.6 in large, full tankers and ore-carrying ships, tests have shown reductions in total resistance up to 15 percent with the new-type bulbs. At such a speed, this nearly approaches total elimination of wave-making resistance. Also, these greater resistance reductions appear to occur on ships of full shapes (block coefficients of 0.8 or more) and pronounced U-sections forward, where the shape includes flat-bottomed sections well forward. The reductions also are greater when the ship is light or higher in the water rather than at full load. This means that bulbs on these ships are more effective nearer the surface. Such phenomena, when considering the added wetted area of the bulbous form with consequent additional frictional resistance, leads to speculation that there is more than a simple reduction of wave-making resistance. Investigators of this type of form at the Uni-

versity of Michigan towing tank have suggested that an alteration in flow characteristics around the bow and along the bottom of the bulbous form is the source of the reduction. Without the bulb, a separation in the flow near and under the bow exists. With the bulb, this flow is improved, and its average velocity over the entire underbody is slowed so that a reduction in viscous resistance occurs (figure 11-10).

M. Kinoshita has pointed out that, in addition to the probable separation phenomenon (which might not always exist), the large bulbous bow extension gradually and uniformly reduces pressure from the bow, aft. Without the bulb this uniformity does not exist, but rather the pressure drops off rather quickly from positive to negative. Such sudden pressure reduction produces an increased velocity with accompanying higher frictional resistance and generally unstable flow characteristics. (For further discussion of bulbous bows, see Kracht 1978).

11-6 Frictional Resistance

The nature of frictional or skin resistance was, to some extent, more evident and familiar to scientists than wave-making resistance before Froude's revealing studies. It



Figure 11-9. Molded bulbous bow on a moderate-sized bulk carrier (launching day, Japan). (Courtesy of Hitachi Zosen.)

was known, for example, that frictional resistance was a function of the surface roughness, the area of the surface, the velocity of the fluid over the surface, and the density of the fluid. There was some confusion concerning the nature of the flow of the fluid, and consequently no meaningful quantitative analysis had been made. Froude was obliged to develop his own means to evaluate frictional resistance, and while these means were workable, they were also awkward and based largely on experimental data where reproducibility was sometimes doubtful. Ultimately, a fuller understanding of frictional resistance emerged with the investigations of Sir Osborne Reynolds around 1883 (Reynolds 1883).

Reynolds conducted a series of experiments on the factors controlling the pressure drop in pipes and conduits. Theoretical predictions based on the work of G. H. L. Hagen in 1839 and J. L. M. Poiseuille in 1840 suggested that the pressure drop should vary linearly with velocity. Actual measurements by civil engineers, including Froude, showed that “the differences of hy-

drostatic pressure at different points (in a pipe) vary as the difference of the squares of the velocities at those points” (Froude 1955, p. 283). Reynolds set up an experiment in which colored dye could be injected into a clear tube fed from a large reservoir to keep the flow relatively constant (figure 11-11). At low velocities, the dye filament remained stable and the flow appeared to move smoothly along streamline layers parallel to the walls with the maximum velocity at the center of the pipe. This was called *laminar flow*. The effects of fluid viscosity were confined to a smooth variation in the velocity across the tube (figure 11-12a).

When Reynolds increased the flow rate through the tube, the dye streaks were observed to become sinuous, then unstable, and finally the whole flow became mixed with unobservable dye streaks. This type of flow he called *turbulent*, and it was characterized by random, large-scale momentum transport and a more uniform mean velocity across the pipe, except next to the wall, where it fell off rapidly (figure 11-12b). By changing the diameters of the tubes, the mean velocity (flow rate), and the fluid kinematic viscosity, he was able to determine that the breakdown of the laminar flow into turbulence appeared to depend on a dimensionless combination of these variables that equaled approximately 2,000, unless great care was taken not to disturb the incoming flow. This combination of variables, now known as the Reynolds number, corresponds to the π term given by equation 11-8. The kinematic viscosity is replaced by its equivalent of absolute viscosity (μ) divided by density (ρ); that is,

$$R_n(\text{critical}) = \frac{vD}{\nu} = \frac{\rho v D}{\mu} \approx 2,000 \quad (11-17)$$

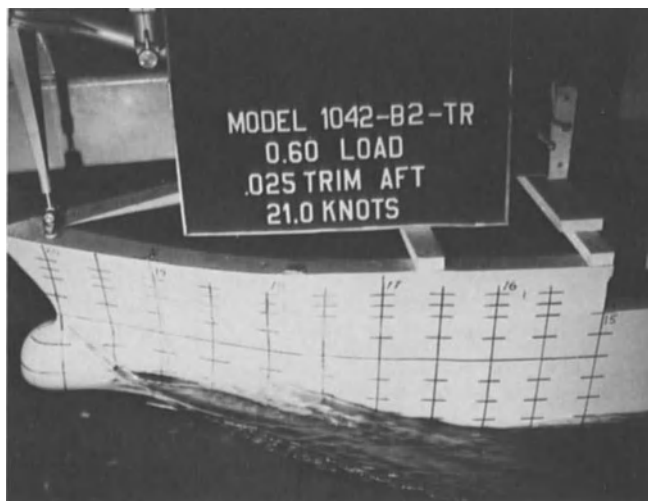


Figure 11-10. Towing tank test of a model with slow-speed bulbous bow near optimum speed. Courtesy of the University of Michigan.)

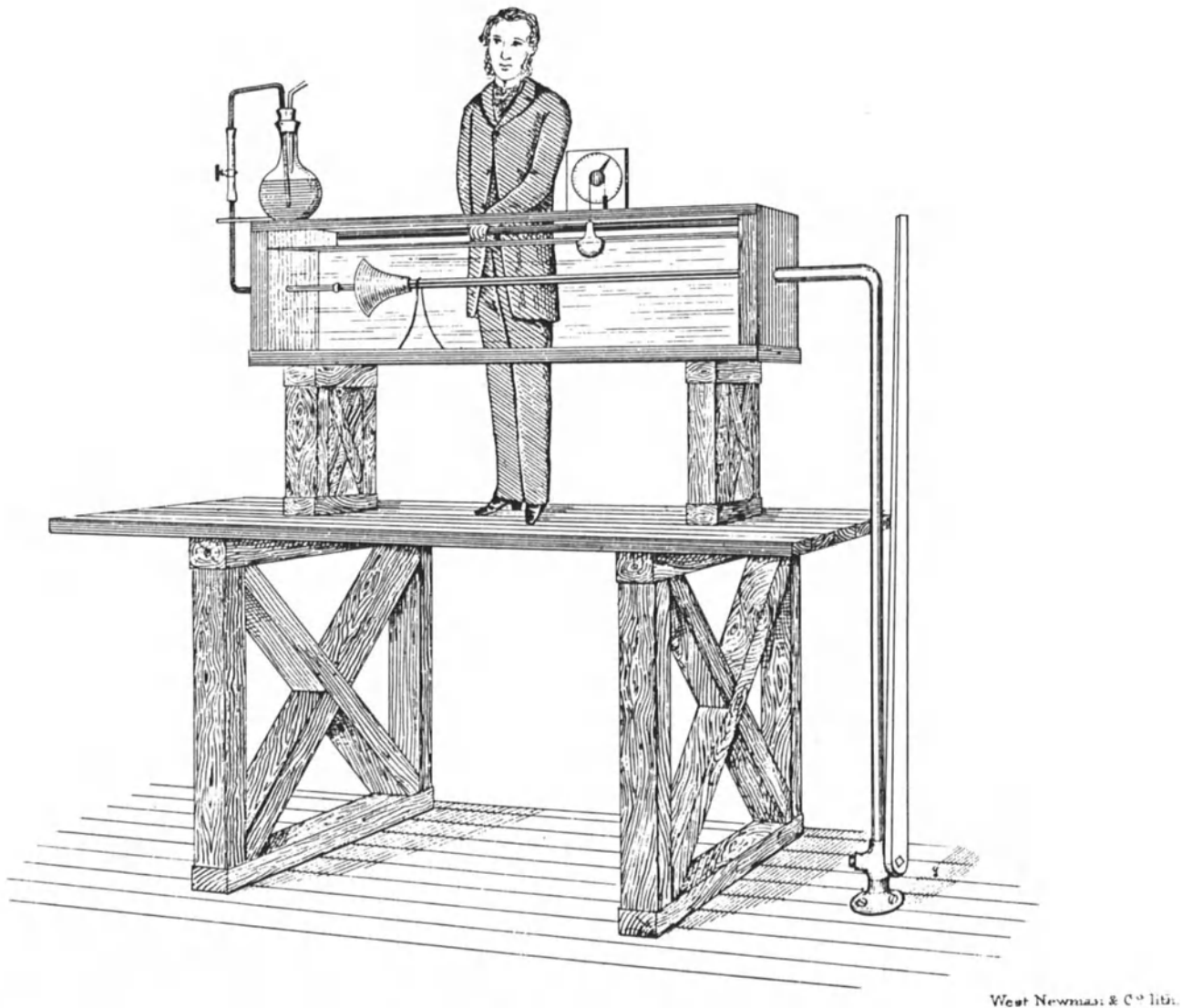


Figure 11-11. Experimental pipe-flow apparatus of Osbourne Reynolds. (From Reynolds 1883.)

where v = average fluid velocity in ft/sec
 D = pipe diameter in ft
 ν = kinematic viscosity of the fluid in ft²/sec

$$= \frac{\mu}{\rho}$$

The critical Reynolds number of about 2,000 defines when transition from laminar to turbulent flow may begin for *internal* pipe flows. The measured pressure

drop for laminar flow does vary linearly with velocity (flow rate) as predicted by the theory of Hagen and Poiseuille. But the measured pressure drop for turbulent pipe flow varies with the square of the velocity for rough pipes and with a power of less than 2 for turbulent flow in a smooth pipe.

The skin friction experiments carried out by Froude and others on planks and flat plates are *external* flows, however, and the necessary change in the characteristic

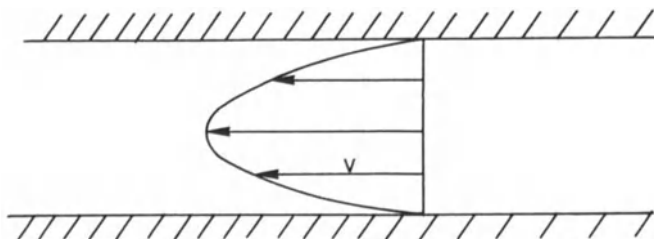


Figure 11-12a. Laminar velocity profile in a pipe

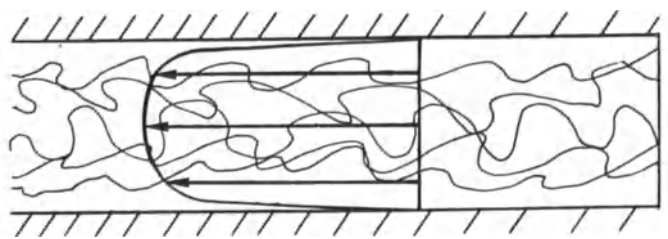


Figure 11-12b. Turbulent velocity profile in a pipe

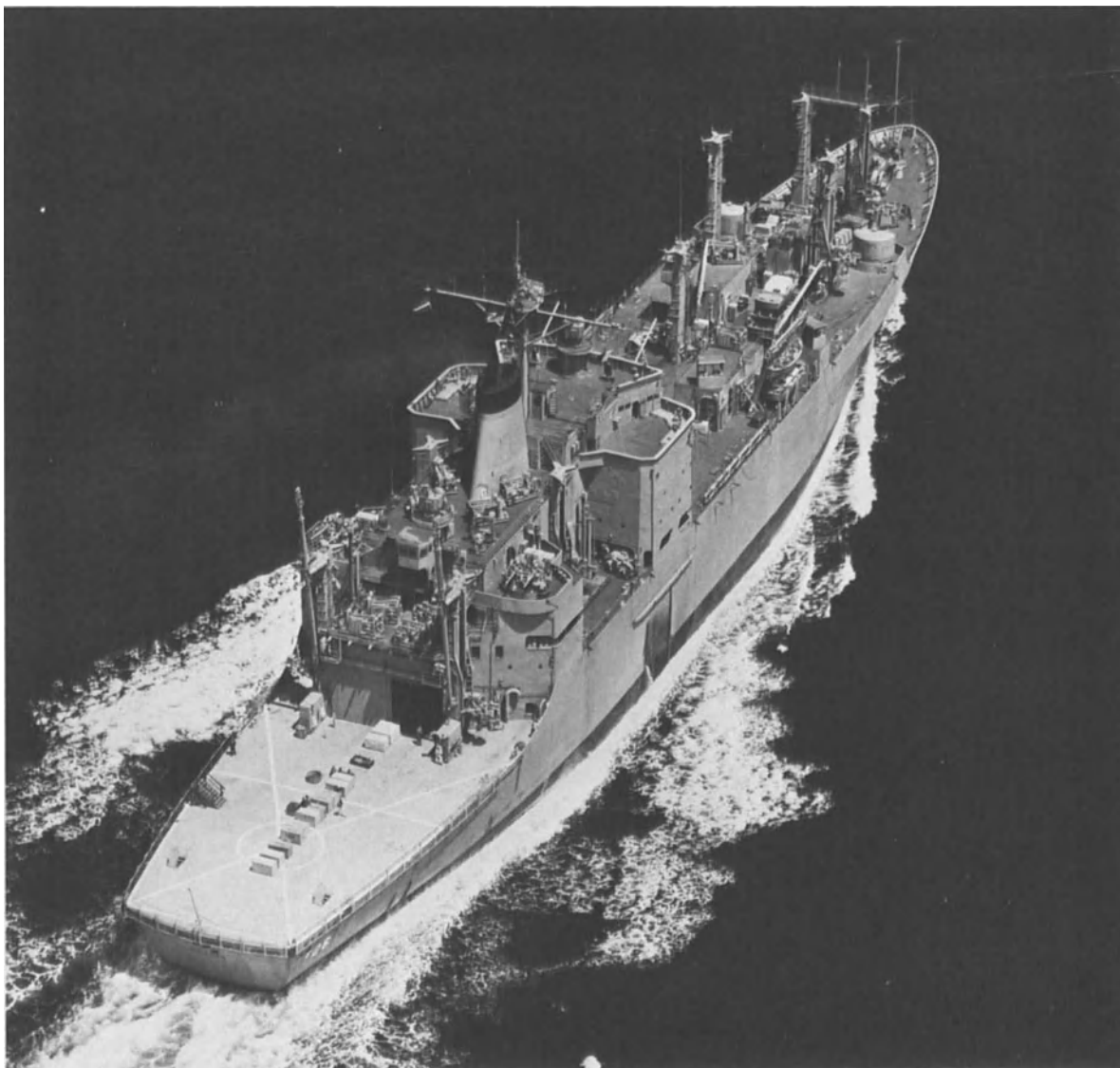


Figure 11-13. The boundary layer shows as a white band adjacent to this ship's hull. The waves angle out in oblique crests at $V = 0.6 \sqrt{L}$. (Courtesy of General Dynamics.)

length in the equation for Reynolds number gives an entirely different critical value for transition.

Early in the twentieth century, G. S. Baker plotted much of the skin-friction data for smooth flat plates in the form of a non-dimensional skin-friction coefficient C_f (equivalent to equation 11-5) versus a Reynolds number based on the length of the plank (equation 11-8), and found good correlation with Froude's results, except at low Reynolds numbers. It is now known that laminar flow is generally found below a length Reynolds number of 4 to 5×10^5 , and turbulent flow is generally found for length Reynolds numbers above 10^6 . Transition is very sensitive to surface roughness and body shape, however, and laminar flow can be maintained to Reynolds numbers above 10^7 using special techniques on *laminar-flow shapes*.

Another way of interpreting Reynolds number is as

the ratio of *inertial forces* to *viscous forces* in a fluid flow. At "low" Reynolds numbers, the viscous forces dominate the flow and it remains laminar, with momentum exchange confined to the molecular level (viscosity). At "high" Reynolds numbers, the inertial forces dominate the turbulent flow with random momentum interchange on a fluid particle (macroscopic) level.

BOUNDARY-LAYER PHENOMENA

The *boundary layer* was defined by L. Prandtl in 1904 as the region in an external flow where the velocity varies from zero at the wall to approximately 99 percent of the free stream (external) velocity. It is the region where the effects of fluid viscosity are present. Anyone who has traveled by ship and has looked down over the side at the passing water has surely noticed the white, foamy region next to the hull (figure 11-13). This is the

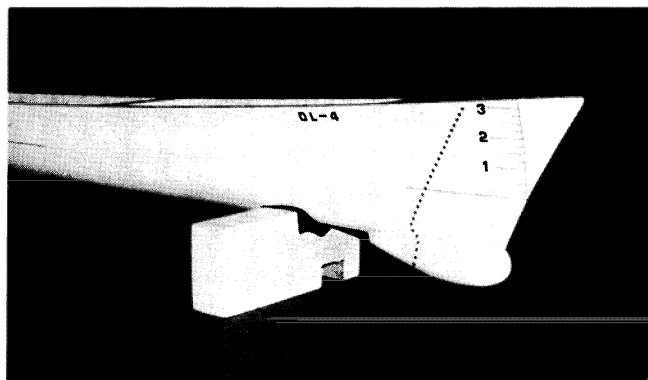


Figure 11-14. Turbulence stimulators (studs) located near the bow of a destroyer model

visible portion of the boundary layer at the surface, and it is convincingly turbulent. Because of the ship's speed, length, and comparative roughness of surface, this boundary layer is always essentially turbulent. Hence all calculations for frictional resistance must recognize that this flow around the model must be artificially triggered into a turbulent condition to maintain a rational comparison (figure 11-14).

Describing the flow pattern about an ideal hull form of smooth surface when towed through calm water will be helpful in understanding better the nature and occurrence of fluid friction. At a slow and steady speed, with F_n less than 0.1, a small hull-form model in a towing

tank in many situations has nearly 100 percent laminar flow about its hull with perhaps only a small area of turbulence near the stern, even if turbulence stimulators are used. If the speed or length or both are increased, the transition point from laminar to turbulent flow moves forward. It will continue to move forward with increasing R_n until most of the flow is turbulent (figure 11-15). However, the transition point does not move proportionately with increased Reynolds number, and this suggests that frictional resistance is also in some way subject to the shape or form of the hull.

The flow lines from the bow, back to the maximum width where inward curvature sets in, are crowded. From this region aft, they open out into a pattern of more expanded and slower flow, and at some point in the after region of the conventional ship form, may actually tend to separate from the hull. Beyond this point of separation, the water is full of swirls and eddies. The drag that results from this separated flow becomes a significant portion of what is called *eddy resistance*.

Historically, eddy resistance has been lumped together with wave-making resistance, simply because it is not a type of frictional resistance whose behavior and effect can be dealt with in a similar way. It is a comparatively small part of total resistance. Since wave-making resistance is a function of form, as eddy resistance obviously is, they are naturally related. However,

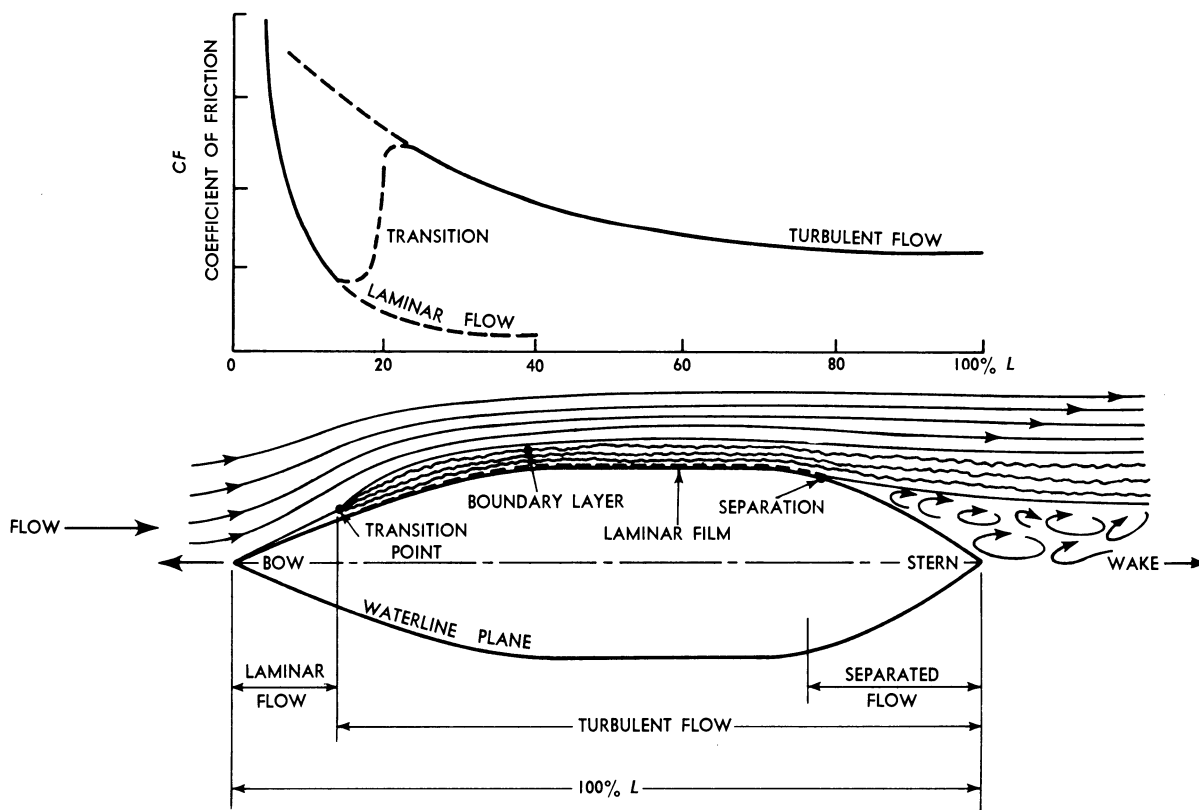


Figure 11-15. Transitional flow of water about a ship model at LWL. The extent of laminar flow and transition to turbulent flow depends on the size of the ship, her speed, the smoothness of the hull, etc. There may be no laminar flow at all in larger vessels.

in the design process they should be treated separately. On the other hand, eddy resistance is most certainly associated with the formation of a stern wave, where they both originate in the low-pressure regions of the stern.

SURFACE ROUGHNESS

As was pointed out in the discussion of pipe flows, skin friction increases with the square of the velocity for rough surfaces, but increases at a power less than 2 for smooth surfaces. The same is true for external flows, so it is important to make a hull as smooth as possible to reduce skin friction. If a surface is *hydraulically smooth*, it is known that there exists a *viscous sublayer* beneath the turbulent boundary layer in which the rate of production of turbulent energy is less than that produced by a rough surface. This means that the resulting local-fluid shear stress at the solid surface is less; therefore, the overall drag of the body will be less. As the Reynolds number increases at a particular location, the viscous sublayer gets thinner and surfaces that are *hydraulically smooth* at low Reynolds numbers may appear as rough surfaces at high Reynolds numbers. Since the turbulent boundary layer and the viscous sublayer both get thicker in going from bow to stern, roughness has less effect at the stern than at the bow. Anti-fouling paints should be as smooth as possible if minimum skin friction is desired. Once a surface is hydraulically smooth, however, there is nothing that can be done to reduce the frictional drag further for a given turbulent boundary layer.

The addition of *polymer additives* is known to change the nature of turbulent flow and reduce the skin-friction drag by as much as 80 percent (Hoyt 1972). The previously mentioned *laminar flow bodies* that depend on shape, heating, or suction to maintain a low-friction laminar-boundary layer must be exceptionally smooth to prevent premature transition.

11-7 Other Resistance Factors

The total resistance opposed to the propulsive force of the ship is mainly composed of the energy in the self-generated waves and in the fluid friction surrounding the ship's immersed and moving hull. Other factors and components of the total resistance, under some conditions, present formidable opposition to forward motion.

Eddy resistance has already been mentioned, and it will be sufficient to add that eddies are produced as a phenomenon of separation in the regular flow pattern around the hull. This separation exists primarily where appendages or sudden changes in the hull's contour are present. Particularly in the after portions of the underbody, which is sometimes referred to as the *run*, excessive curvature located in an expanding flow pattern

inevitably produces a breakaway of the flow. The resulting eddy resistance exists together with the wave-making resistance of the stern wave, which forms in the same region of lowered pressure. These are two different components, to be dealt with separately, but their magnitudes have the same or similar causes.

Air resistance becomes a factor of increasing importance in higher-speed ships of proportionately large superstructures. Attention is being paid more consistently in modern ships of all sizes to the shape and fairing of superstructure, stacks, masts, etc. Wind-tunnel tests of waterline models are most helpful in evaluating this reduced wind resistance. With speeds in excess of 20 knots, it is economical to fair the superstructure and optimize its air-drag character.

Shallow water produces resistance problems that become theoretically effective at depths equal to $10T(V/\sqrt{L})$, where T is the vessel's mean draft. The shallow-water problem becomes far more noticeable with depths that exist in many harbors and particularly canals. Between the bottom of a canal and the bottom of a ship, the channeling effect of the water must increase its velocity, at the same time decreasing its pressure. This produces increased friction, due to both higher flow rate and sinkage that increases the wetted surface. Such restrictions produce a blockage effect that in turn changes the character of the self-generated wave. This situation is difficult to define or evaluate because of the infinite combinations of water depth, vessel size, etc. However, there is a limiting relation of speed to water depth alone, where the shallow water effect increases resistance due to the changing wave-making phenomena. This is

$$\frac{v}{\sqrt{gh}} = 1.0$$

where h = water depth.

Where the increased drag is due to the shallow-water effect alone, it will finally begin to decrease if the ship has sufficient power and the appropriate shape to overcome it.

Rough water, most powerful and erratic of all the factors that oppose the propulsive force, is the most difficult to analyze. The action of rough water cannot be predicted in a deterministic way. Not surprisingly, however, it can be dealt with in analysis by computers.

The nature of sea waves and sea conditions will be discussed in chapter 13. It will be sufficient to note here the manner in which the ship's resistance is increased in rough water and, for the moment, to avoid the complexities of controlling her motion. Actually, sea waves have two effects in increasing resistance. The first is a function simply of their size and the force of moving water that acts against the ship's forward motion. The

second effect is the motion they induce in the ship, either pitching, rolling, or heaving, that adds to the wetted area as well as the localized increased water volume to be displaced. In the design process, both of these effects are considered in the light of consequent functional limitations. Obviously, little attention is given to the matter in the design of, say, a harbor ferry or river barge. Small protected-water vessels are known to have their limitations. However, in the design of seagoing craft, the means of providing for the action of the sea are not unlimited. As a matter of fact, assuming the strength factors are attended to, the necessary forms, which include a fine and easy entrance with adequate sheer, are difficult to describe geometrically in terms of seaworthiness. However, such forms as fishing trawlers are universally recognized for their advantages in heavy seas. Unfortunately, when compared in calm or moderate seas with certain conventional ship forms of a greater length-beam ratio and generally less curvature, they do not perform as well.

Such considerations point up the most fundamental of design principles: any single ship concept must always be a compromise. No ship can be designed for all things; a ship's features must answer those functions she is to perform and the probable environment she will encounter.

11-8 Effective Horsepower

The concept of *effective horsepower (EHP)* as the power due to the ship's resistance was suggested by William Froude in 1876 (Froude 1955, p. 289). He was searching for a reference power with which to measure the efficiency of the propulsion system. Froude defined EHP as

$$\frac{\text{ship resistance} \times \text{speed in ft/min}}{33,000 \text{ ft/min/hp}}$$

since power is force times velocity converted to horsepower. In this text, velocity is usually given in feet per second (fps), so the equation for effective horsepower is

$$\text{EHP} = \frac{R_T(\text{lb})v(\text{fps})}{550 \text{ ft-lb/sec/hp}} \quad (11-18)$$

If equation 11-6 for total resistance is substituted in equation 11-18,

$$\text{EHP} = C_T \frac{1}{2} \rho v^3 S / 550 \quad (11-19)$$

so that EHP is seen to vary as C_T times the cube of the velocity for a given ship. At low Froude numbers, frictional resistance dominates and C_T decreases with increasing speed, so the EHP varies with speed to a power less than 3. As a ship approaches hull speed, C_T rises

rapidly and the EHP may vary with speed to a power approaching 6 or 7.

11-9 Current Practice in EHP Model Testing

The International Towing Tank Conference (ITTC) was organized in 1932-33 as an International Conference of Ship Tank Superintendents to promote the exchange of knowledge and experience between ship hydrodynamic laboratories. At present, there are over seventy member organizations from all parts of the world who participate in the general meetings held every three years. Between conferences, technical committees of the ITTC review developments in ship hydrodynamics and evaluate proposed new methods for improving predictions of ship performance based on model tests. The committees make recommendations to the general conference for the standardization of model-test procedures and for the analysis and exchange of data concerning test results. Despite many attempts to improve upon W. Froude's method for extrapolating model-test data to full-scale resistance predictions, his method still remains intact, except for the use of dimensionless coefficients and parameters, and improvements in his formulation for estimating skin-friction resistance.

VISCOUS (FRICTIONAL) RESISTANCE

As mentioned previously, G. S. Baker introduced the use of dimensionless numbers and coefficients in calculating frictional resistance from

$$R_F = C_F \frac{1}{2} \rho v^2 S \quad (11-20)$$

as suggested by equation 11-5. He also plotted C_F values versus Reynolds number in analyzing the data from many plank experiments. In 1932 K. Schoenherr developed an empirical formula to fit the data for turbulent boundary layers along a smooth flat plate.

$$\frac{0.242}{\sqrt{C_F}} = \log_{10}(R_n C_F) \quad (11-21)$$

This formulation was adopted by the American Towing Tank Conference (ATTC) in 1947 as a proposed standard method for calculating equivalent flat-plate resistance for both model and ship. It was also proposed that an allowance of .0004 for clean, new vessels be added to the sum of the frictional- and residual-resistance coefficients for the purpose of predicting the total resistance of new ships. Although originally proposed as an allowance for the effect of hull roughness on ship resistance, the factor was later adopted as a model-ship correlation allowance (C_A) by the ITTC to account for various differences between the ship resistance as deduced from full-scale trials and that predicted from model

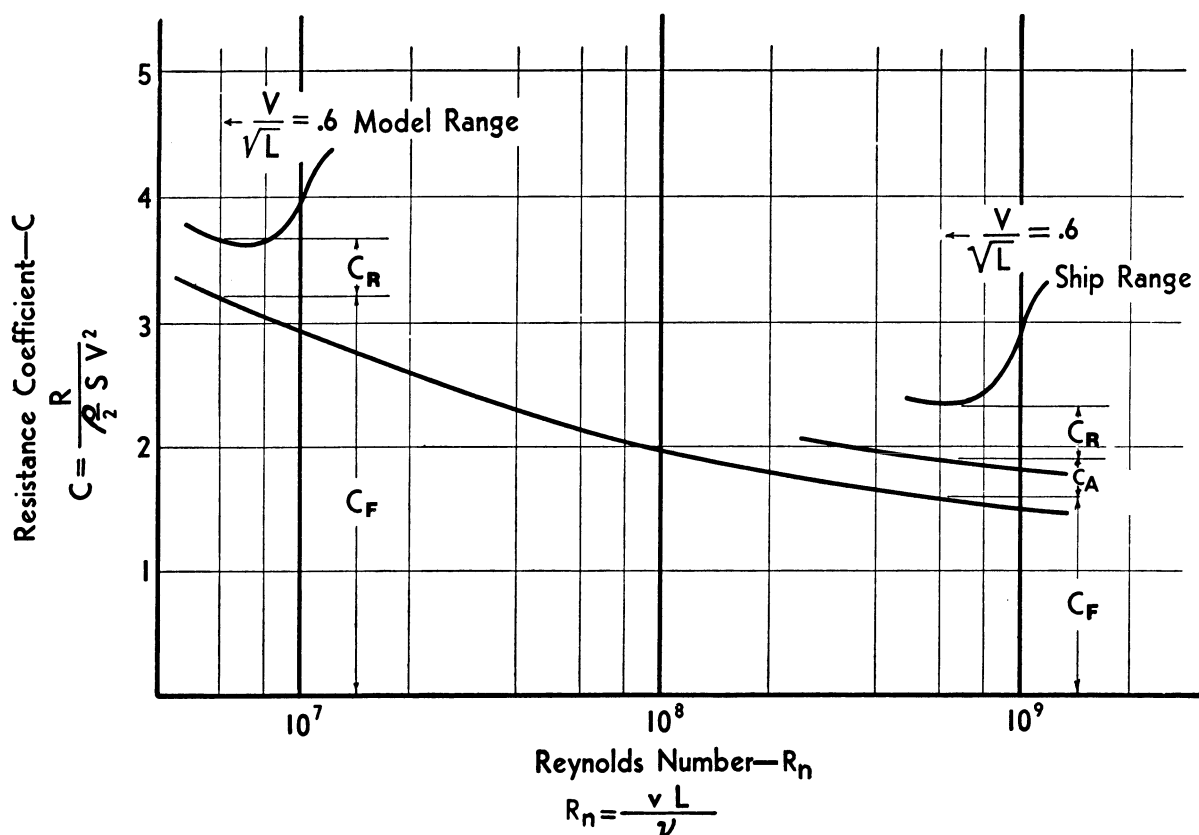


Figure 11-16. Coefficients of resistance versus Reynolds number

tests. That is,

$$C_{Ts} = C_{Fs}(R_n) + C_{RM}(F_n) + C_A \quad (11-22)$$

since $C_{Rs}(F_n)$ is assumed to equal $C_{RM}(F_n)$ for equal Froude numbers.

The *Schoenherr line* formula (equation 11-21) underestimated the frictional resistance of small models that must use turbulence stimulators to induce transition from laminar to turbulent boundary-layer flow at low Reynolds numbers. Also, equation 11-21 cannot be solved explicitly for C_F as a function of Reynolds number. In 1957 the ITTC adopted its 1957 model-ship correlation line as an interim solution for the prediction of ship resistance from model tests. It is still in use today as

$$C_F = \frac{.075}{(\log_{10} R_n - 2)^2} \quad (11-23)$$

Equation 11-23 does *not* represent the actual frictional resistance of an equivalent flat plate at model-scale Reynolds numbers. It attempts to account for the viscous resistance of relatively thin model ships of varying sizes as well as the frictional resistance of the full-scale ship. Equations 11-21 and 11-23 give essentially the same values of C_F above Reynolds numbers of 10^7 , and either may be used with equation 11-22 to predict the total-resistance coefficient of the ship.

If the frictional resistance of a model were equal to that of the equivalent flat plate, the total-resistance coefficient C_T would approach the frictional resistance coefficient C_F as the wave-making resistance became vanishingly small at low Froude numbers. This is nearly so for *thin ships* such as destroyers and other high-speed displacement ships. However, so-called *full-form* or high-block-coefficient ships exhibit apparent residual-resistance coefficients that do not approach zero at low Froude numbers (figure 11-16). G. Hughes and others have proposed that a form factor k be defined such that

$$C_v(R_n) = (1 + k)C_F(R_n)$$

where C_v = coefficient of viscous resistance to be substituted for C_F .

This approach attempts to account for all the effects of fluid viscosity but suffers from the fact that the “viscous resistance may vary significantly with Froude number (which affects the flow at the edge of the boundary layer) as well as with Reynolds number, so that it is not well predicted on a constant form factor basis” (Oosterveld 1978, p. 21). There are numerous measurement errors associated with determining the low-Froude-number form factor, so another approach is to measure the resistance of a fully submerged “equivalent body of revolution.” The measured resistance of an axisymmetric body is compared with that predicted on the basis

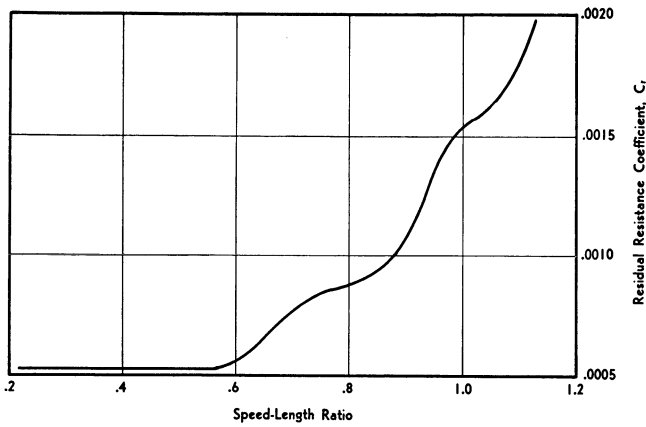


Figure 11-17. Curve of residual resistance coefficient versus speed-length ratio

of an equivalent flat plate. The following formula for the viscous resistance of a body of revolution relative to the Schoenherr line has been suggested by N. Scholz:

$$\frac{C_v}{C_F} = 1 + 0.5 \frac{d}{L} + 3 \left(\frac{d}{L} \right)^3 \quad (11-24)$$

where d = the maximum diameter of the body.

Another formulation for the viscous resistance of surface ships has been suggested by P. S. Granville.

$$\frac{C_v}{C_F} = 1 + 33.8 C_B^2 \left(\frac{B}{L} \right)^2 \left(\frac{T}{B} \right) \quad (11-25)$$

again based on the Schoenherr formulation. However, the performance committee report in the 1978 *Proceedings* of the fifteenth ITTC states that “the problem of predicting the excess of the total viscous resistance of a given hull form over the flat-plate frictional resistance remains unsolved” (Oosterveld 1978, p. 50).

RESIDUAL RESISTANCE

Since the residual (residuary) resistance of a ship includes wave-making, eddy, and wave-breaking (near the bow of full-form ships, as illustrated in figure 11-5) components, there have been theoretical and experimental attempts to separate these components by measuring the wave geometry (wave pattern) of the transverse wave system and also the turbulent wake associated with the eddy resistance. Again quoting from the 1978 *Proceedings* of the ITTC, “The original hopes of employing wave pattern and wake traverse measurement for improving on Froude’s method of scaling ship resistance have not yet materialized” (Oosterveld 1978, p. 25).

High-block-coefficient (full-form) ships have large eddy-resistance components and their residual-resistance coefficients have many “humps and hollows,” as illustrated in figure 11-17. It is still unresolved whether or not the previously mentioned residual resistance off-

set at low Froude numbers (speed-length ratios) can be properly scaled by the use of the correlation allowance that is assumed to be independent of Froude and Reynolds numbers. When actual correlation studies are made, there is considerable variation in the correlation allowance, and negative correlation allowances are often found for very long ships.

This lack of consistency in C_A can be also attributed to the erratic behavior of, or lack of suitable theory for determination of, the phenomena existing in and around the boundary layer in the actual ship. Some of these factors that do not reproduce well in the model might well be found in the eddy resistance, separation resistance, or viscous-pressure drag. The whole problem, while it does not affect the validity of model testing or the reliability of the powering results within reasonable tolerance, is a most challenging one for the serious research student.

APPENDAGE RESISTANCE

Most preliminary model testing and often entire model tests are carried out on bare hull models. This means models are used that do not include such real ship appendages as rudders, bilge keels, propellers, struts, shafts, or bossings. Obviously, such appendages contribute to the total resistance and required power, and complete model tests will determine both the total resistance values with and without appendages. Many model tanks handle the appendage resistance simply as a percentage increase based on the ratio of the bare hull resistance to the hull resistance with appendages. The percentage on the basis of model tests, because of the variation of scaling effects in eddy and viscous-pressure drags, is sometimes modified by a factor of 2 when applied to the ship’s resistance. There are few rational methods to accurately determine appendage resistance other than full-scale testing, which is most often impractical. In any case, it must be considered and included for the final determination of EHP and SHP (see chapter 12).

MODEL SIZE

It has been found through experience that it is difficult to establish a turbulent boundary layer at low speeds on models less than 5 feet (1.5 meters) in length. Even with stud stimulation, the flow will be dominated by irregular transition below Froude numbers of 0.1. The lower limit on model size for EHP testing is also related to the relative importance of the wave-making, eddy-making, and skin-friction resistance in the principal speed range of the ship. High-speed ships (F_n greater than 0.35) generally have fine forms, low block coefficients, and transom sterns. Wave-making resistance dominates at high speed. Eddy-making resistance is rather independent of model size, since the separation point at the transom stern is the same for model and full-scale ships.

Thus small models can be used very effectively and economically during the preliminary design stage to assist the hull-form designer in selecting the best hull form for his mission requirements.

Large merchant ships such as supertankers and bulk carriers are very full formed, have high block coefficients, and have sharply rounded sterns and bulbous bows. They are designed to travel at low Froude numbers and have low ratios of wave-making to skin-friction resistance. The eddy resistance, because of bow and stern separation and vortex formation at the bow, is very Reynolds-number dependent, so there has been a trend to use the largest possible model a towing tank can handle before *blockage* (related to the ratio of the model cross section to the tank cross section) becomes a problem. Large towing tanks may use models up to 13 to 14 meters (42 to 46 feet) in length in order to avoid having severe stern-separation problems on the models. This separation probably does not occur at the very large Reynolds numbers involved in full-scale ships.

The relationship between model size and Reynolds number for a given Froude number is found from

$$R_{nM} = \frac{v_M L_M}{\nu_M} \quad \text{and} \quad v_M = v_S \left(\frac{L_M}{L_S} \right)^{1/2}$$

Thus,

$$R_{nM} = \frac{v_S}{\sqrt{L_S}} \frac{(L_M)^{3/2}}{\nu_M} \quad (11-26)$$

If the model size is increased by a factor of 4, the model Reynolds number at a given speed-length ratio is increased by a factor of 8. The proper choice of model size for EHP testing of a particular hull-form type becomes a trade off between the lower cost of small models, tested in small towing tanks, and the better accuracy of more expensive large models of full-form ships, tested at higher Reynolds numbers in large towing tanks.

The testing of models fitted with propellers for shaft horsepower (*SHP*) tests involving torque and thrust measurements requires that the model scale be selected by the propeller size needed to ensure adequate Reynolds numbers on the propeller. This problem is discussed in the next chapter.

TEST PROCEDURE

To illustrate the procedure for estimating full-scale ship EHP characteristics, data from actual model tests of the USCG cutter *Hamilton* will be used in a step-by-step analysis on a hand calculator. This analysis will then be compared with the computer-generated model test data and the full-scale (prototype) expansion of thirty-six test runs at various speeds in the 120-ft towing tank at the U.S. Naval Academy.

The *Hamilton*-class cutters are 350 feet long on the

waterline, have a displacement of 2,720 tons, and a wetted surface of 15,797 square feet at the test conditions. The model data will be expanded to standard seawater at 59°F, which has a density of 1.9905 slugs/ft³ and a kinematic viscosity of 1.2791×10^{-5} ft²/sec.

The scale model has a waterline length of 5.5 feet. It was tested in the 120-ft towing tank in water at 69°F with a density of 1.9365 slugs/ft³ and a kinematic viscosity of 1.066×10^{-5} ft²/sec. The linear scale ratio of the dimensions is therefore

$$\lambda = \frac{350}{5.5} = 63.636$$

The model wetted surface varies as the inverse of the scale ratio squared.

$$\begin{aligned} S_M &= \frac{S_S}{\lambda^2} \\ &= \frac{15,797}{63.636^2} = 3.90 \text{ ft}^2 \end{aligned}$$

The model must be ballasted to the proper displacement.

$$\begin{aligned} \nabla_M &= \frac{\nabla_S}{\lambda^3} \\ \Delta_M &= \frac{\rho_M}{\rho_S} \frac{\Delta_S}{\lambda^3} \\ \Delta_M &= \frac{1.9365}{1.9905} \frac{2,720 \text{ tons}}{(63.636)^3} \times 2,240 \frac{\text{lb}}{\text{ton}} \\ &= 23.00 \text{ lb} \end{aligned}$$

The *Hamilton*-class cutter is a relatively fine-form ship, with a block coefficient of 0.49, a length-to-beam ratio of 8.33, and a beam-to-draft ratio of 3.11. A check of equation 11-25 for form factor gives a value of .038, which is quite low. Consequently, form factors will be ignored in this analysis, and a standard correlation allowance of .0004 will be applied to the ship expansion.

One of the test runs at a model speed of 5.55 ft/sec (3.29 kt) gave a total model resistance of 0.845 lb. This corresponds to a Froude number of 0.42 or a $V/\sqrt{L}$ of 1.4. The following steps illustrate the procedure, which is repeated for each of the test runs.

1. Calculate the total resistance coefficient.

$$\begin{aligned} C_{TM} &= \frac{R_{TM}}{\frac{1}{2} \rho_M v_M^2 S_M} \\ &= \frac{0.845}{\frac{1.9365}{2} (5.55)^2 3.9} \\ &= 7.265 \times 10^{-3} \end{aligned}$$

2. Calculate the model Reynolds number.

$$\begin{aligned} R_{nM} &= \frac{v_M L_M}{\nu_M} \\ &= \frac{5.55(5.5)}{1.066 \times 10^{-5}} \\ &= 2.864 \times 10^6 \end{aligned}$$

3. Calculate the frictional-resistance coefficient of the model, using the 1957 ITTC line.

$$\begin{aligned} C_{FM} &= \frac{.075}{(\log_{10} R_n - 2)^2} \\ &= 3.776 \times 10^{-3} \end{aligned}$$

4. Calculate the residual-resistance coefficient of the model and ship at the same Froude number.

$$\begin{aligned} C_{RS} &= C_{RM} = C_{TM} - C_{FM} \\ &= (7.265 - 3.776) \times 10^{-3} \\ &= 3.489 \times 10^{-3} \end{aligned}$$

5. Calculate the ship Reynolds number.

$$\begin{aligned} v_s &= v_M \sqrt{\lambda} \\ &= 5.55 \sqrt{63.636} = 44.27 \text{ ft/sec} \\ &= 26.24 \text{ kt} \\ R_{ns} &= \frac{v_s L_s}{\nu_s} = \frac{44.27(350)}{1.2791 \times 10^{-5}} \\ &= 1.21 \times 10^9 \end{aligned}$$

6. Calculate the frictional-resistance coefficient for the ship, using the 1957 ITTC line.

$$\begin{aligned} C_{Fs} &= \frac{.075}{(\log_{10} R_n - 2)^2} \\ &= 1.49 \times 10^{-3} \end{aligned}$$

7. Calculate the total-resistance coefficient for the ship, using a correlation allowance of .0004.

$$\begin{aligned} C_{Ts} &= C_{Rs} + C_{Fs} + C_A \\ &= 5.38 \times 10^{-3} \end{aligned}$$

8. Calculate the total resistance of the ship at the corresponding Froude number.

$$\begin{aligned} R_{Ts} &= C_{Ts} \frac{\rho_s}{2} v_s^2 S_s \\ &= 5.38 \times 10^{-3} \frac{1.9905}{2} (44.27)^2 15,797 \\ &= 1.659 \times 10^5 \text{ lb} \end{aligned}$$

9. Calculate the EHP of the ship at a speed of 44.27 ft/sec.

$$\begin{aligned} \text{EHP} &= \frac{R_{Ts} v_s}{550} = \frac{1.659 \times 10^5 (44.27)}{550} \\ &= 13,350 \text{ hp} \\ &\approx \boxed{13,400 \text{ horsepower}} \end{aligned}$$

Note that more significant figures than are justified by the data were carried throughout the calculation in order to reduce round-off errors. The final result is reasonably accurate to three significant figures.

These steps are repeated for each of the model test runs. A computer plot of the results is shown in figure 11-18. Several things should be noted in figure 11-18. There is little data scatter for this 5.5-ft-long model of transom-stern ship above a Froude number of 0.1. Below 0.1, however, there is considerable data scatter and the values of the residual-resistance coefficient below zero indicate the presence of a partially laminar boundary layer. This data, therefore, is not valid and should be ignored. Since the residual resistance tended to approach zero at low Froude number, ignoring a form-factor correction seems justified.

At the model scale, the wave-making resistance is consistently below the frictional resistance until speeds above the *hull speed* (an F_n of 0.4) are tested. Note that the maximum rate of increase of the total- and residual-resistance coefficients corresponds to the hull speed.

Figure 11-19 illustrates the expansion of the unfaired model test results to full-scale resistance coefficients. If we ignore the data below F_n at 0.1, the lower frictional-resistance coefficients at full scale combined with the corresponding residual-resistance coefficients result in lower total-resistance coefficients throughout the speed range. Note that wave-making resistance begins to dominate the total resistance below hull speed and represents about two-thirds of the total resistance at high Froude numbers around 0.5. The resulting EHP curve is plotted against ship speed in figure 11-20, and the sinkage and trim of the ship are illustrated in figure 11-21. Note that the bow and stern sink together up to 22 knots. The bow rises sharply and the stern sinks further as the hull speed of 25 knots is exceeded.

ESTIMATE OF EHP FROM STANDARD SERIES

In the earlier stages of the design process, when it is not feasible to define the hull form or when designers do not have model-testing facilities readily available, it is possible to make reasonably accurate power calculations by using preplotted model series data. The oldest and most traditional of such sources is Taylor's Standard Series (Comstock 1967, p. 347). This data, originally

USNA HYDROMECHANICS LABORATORY
MODEL TEST RESULTS

TEST NO.: 143
TEST TYPE: EHP, SW, 120 PC

START DATE: 25-OCT-78 09:02:26
LOG DATE: 21-APR-79 10:25:34

MODEL NAME: HAML
DISPLACEMENT: 23.0 LBS
WET SURFACE: 3.9 SQ FT
WATER TEMP: 68.9 F

LWL: 5.50 FT
LCG: -0.099 FT
STIM: STUDS

LEGEND: $\boxplus$ = CTM $\oplus$ = CFM $\oplus$ = CRM

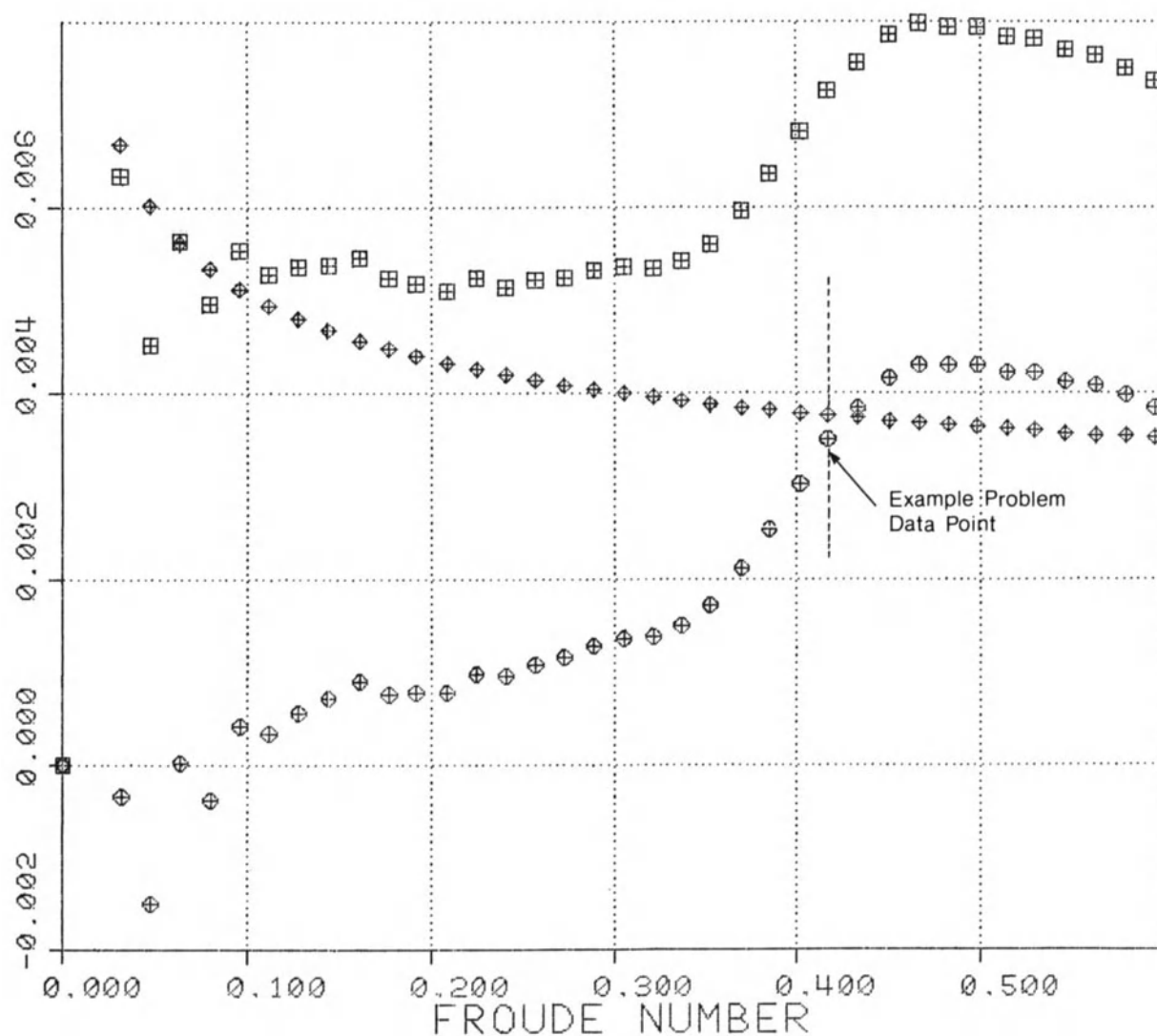


Figure 11-18. Model test results

USNA HYDROMECHANICS LABORATORY
PROTOTYPE SHIP EXPANSION

TEST NO.: 143
TEST TYPE: EHP, SW , 120 PC

START DATE: 25-OCT-78 09:02:26
LOG DATE: 08-NOV-78 15:02:14

MODEL NAME: HAML
DISPLACEMENT: 2719.7 LT SW
WET SURFACE: 15797.3 SQ FT
SCALE RATIO: 63.64

LWL: 350.00 FT
LCG: -6.300 FT
CORRELATION: 0.0004 ITTC
WATER: 59.0 (F) , SW

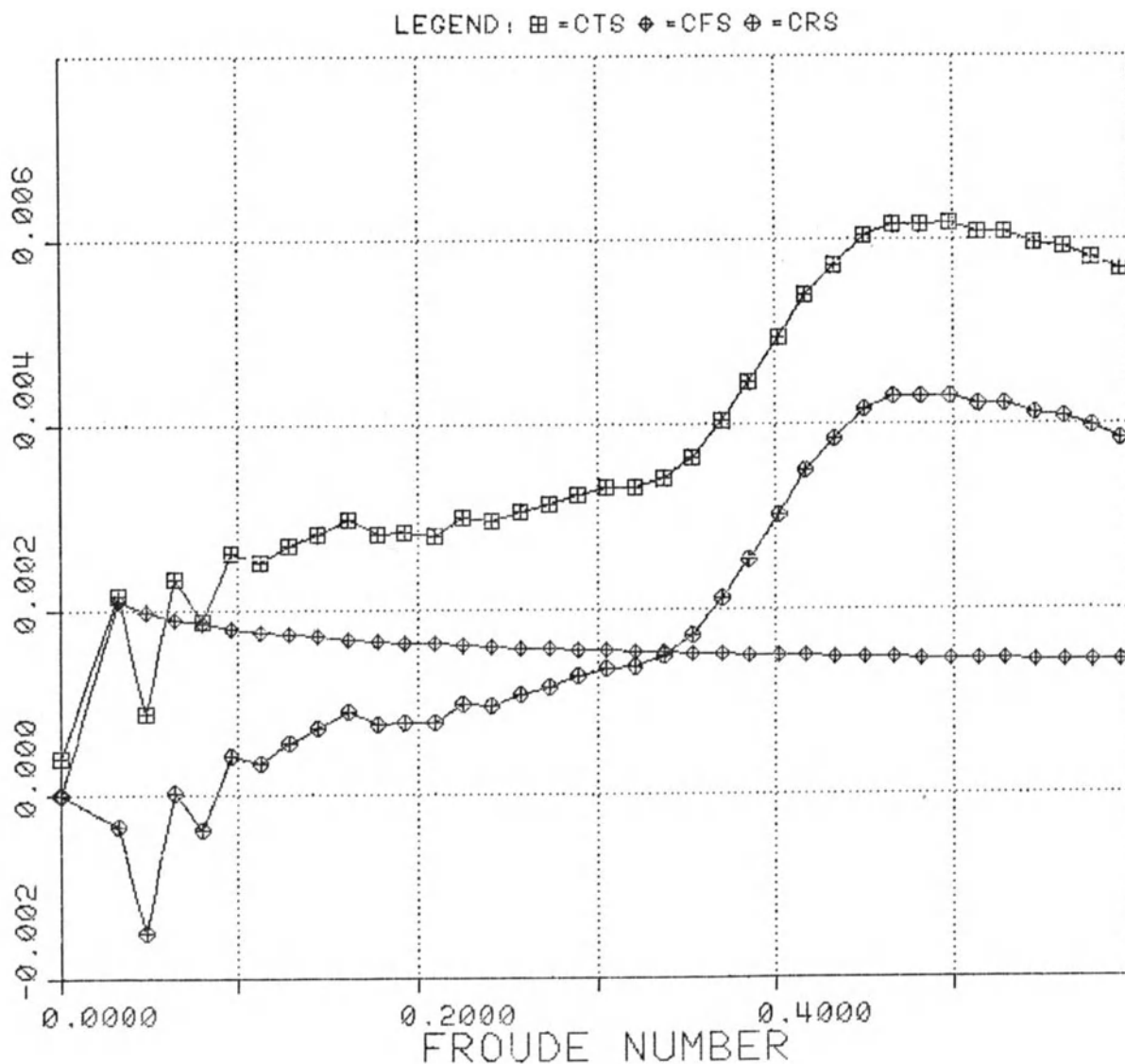


Figure 11-19. Prototype ship expansion

USNA HYDROMECHANICS LABORATORY
PROTOTYPE SHIP EXPANSION

TEST NO.:	143	START DATE:	25-OCT-78 09:02:26
TEST TYPE:	EHP, SW , 120 PC	LOG DATE:	08-NOV-78 15:02:14
MODEL NAME: HAML		LWL:	350.00 FT
DISPLACEMENT:	2719.7 LT SW	LCG:	-6.300 FT
WET SURFACE:	15797.3 SQ FT	CORRELATION:	0.0004 ITTC
SCALE RATIO:	63.64	WATER:	59.0 (F) , SW

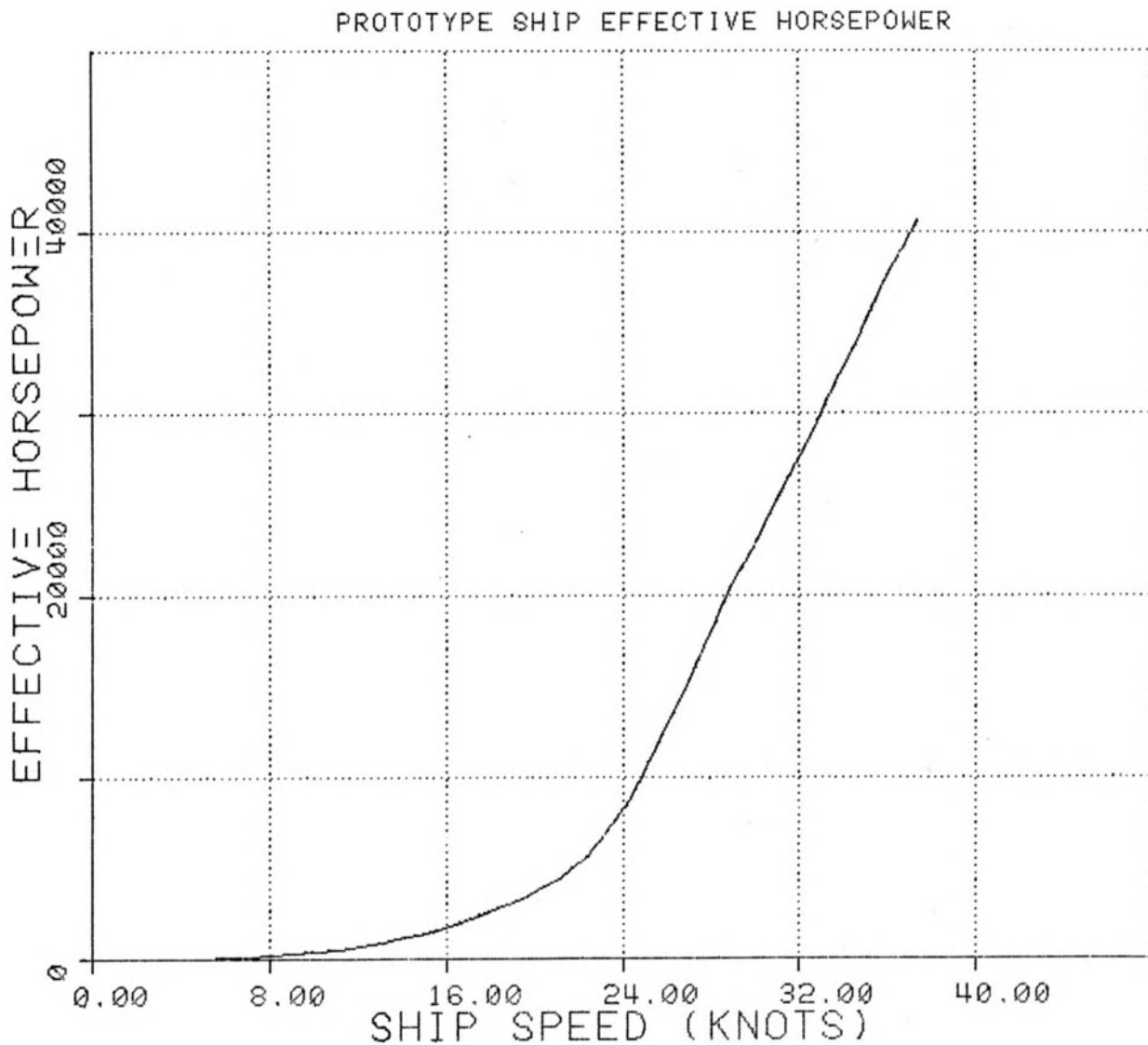


Figure 11-20. Prototype EHP

USNA HYDROMECHANICS LABORATORY
 PROTOTYPE SHIP EXPANSION

TEST NO.: 143
 TEST TYPE: EHP, SW, 120 PC

START DATE: 25-OCT-78 09:02:26
 LOG DATE: 08-NOV-78 15:02:14

MODEL NAME: HAML
 DISPLACEMENT: 2719.7 LT SW
 WET SURFACE: 15797.3 SQ FT
 SCALE RATIO: 69.64

LWL: 350.00 FT
 LCG: -6.300 FT
 CORRELATION: 0.0004 ITTC
 WATER: 59.0 (F), SW

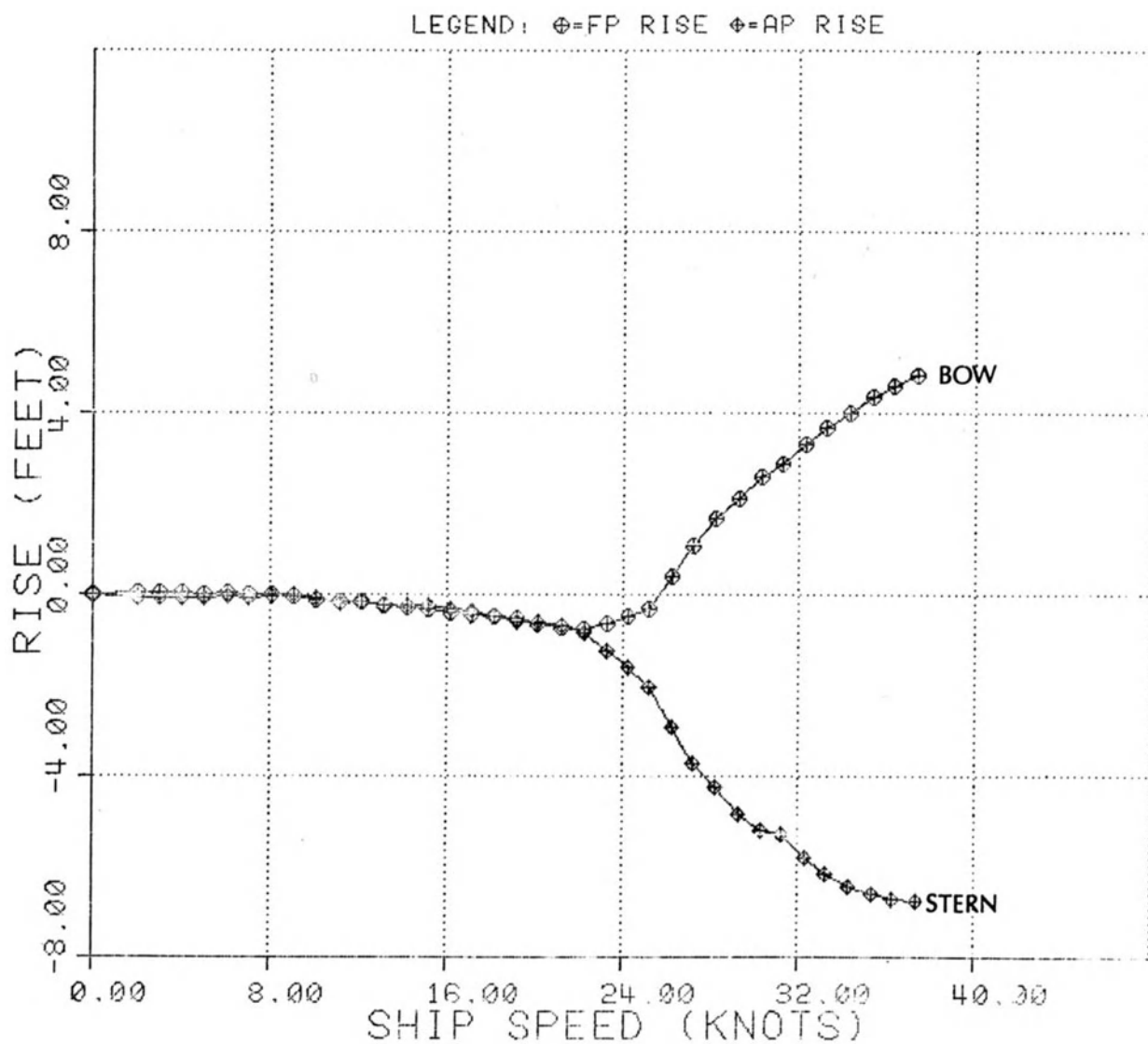


Figure 11-21. Prototype sinkage and trim

compiled and arranged by Admiral David W. Taylor in the early part of this century, was based on model tests of a series of uniformly varied models of similar geometry. It has since become a classic reference tool and has set a pattern for other similar series data. The original Taylor's Series was revised in form in 1954 to bring the data into closer accord with more current methods of computation. It is presently available under the title *A Reanalysis of the Original Test Data for the Taylor Standard Series* (Gertler 1954). It should be understood that the parent model on which this data is based was a hull form with the old-style cruiser stern, which is a sharp-stern form. This form undoubtedly produces a variation in residual resistances in the upper speed ranges from the more contemporary naval hull forms with transom-type sterns. There are, however, series test data available on hulls with transom sterns, as well as trawler hull forms and other more modern hull forms. The Taylor Standard Series data still serve very well, however, for a great many types of cargo ship forms and conventional moderate-speed ships. From these series data, the designer can obtain the residual-resistance coefficient C_R for desired speeds when entering with beam-draft ratio, prismatic coefficient, and volumetric coefficient. His computations for EHP, then, follow the procedure presented in the preceding section and are dependent for their validity on the series value of C_R .

These preliminary power determinations from series data are normally made during the feasibility-study phase of the design and are particularly helpful in determining power requirements in connection with cruising range, cost analysis, and performance requirements.

Problems

11-1. The period T of a pendulum is thought to be a function of the length of the pendulum l , the acceleration of gravity a , and the mass of the pendulum bob m . Using dimensional analysis, establish a dimensionless parameter relating the principal variables.

11-2. Cavitation inception is known to depend on the pressure differential between the local pressure and the vapor pressure of water. Include this pressure difference and rework example 11-1 with seven variables.

11-3. Standing on the stern of a yard patrol craft, you observe that the transverse wave trailing the vessel appears to have a length of 50 feet. Estimate the speed of the vessel in knots.

11-4. Compare the hull speeds of several destroyers having waterline lengths of 383 feet, 408 feet, 415 feet, and 529 feet with that of an aircraft carrier having a waterline length of 990 feet.

11-5. Each of the destroyers mentioned in problem

11-4 is acting as a plane guard for the aircraft carrier that is launching aircraft while traveling at 32 knots. Calculate the Froude number of each of the vessels at this speed.

11-6. Rework the example problem concerning a model test of the USCG-cutter *Hamilton*. At a model speed of 1.644 knots, the measured total resistance was 0.148 pounds. Compare your results with the plotted results in figures 11-17, 11-18, and 11-19.

11-7. Rework the example problem concerning a model test of the USCG-cutter *Hamilton*. At a model speed of 3.670 knots, the measured total resistance was 1.155 pounds. Compare your results with the plotted results in figures 11-17, 11-18, and 11-19.

11-8. Write a computer program to solve problems 11-6, 11-7, and any other set of towing-tank test data based on model and ship characteristics.

11-9. It is desired to run a model test of an FFG7-class frigate in the 380-ft towing tank at the U.S. Naval Academy. The ship is 408 feet long on the waterline, has a wetted surface area of 19,456 square feet, and displaces 3,600 tons in salt water with a density of 1.991 slugs/cu ft.

- a) If the model is built to a scale ratio of 25, determine the waterline length, wetted surface area, and displacement of the model in fresh water with a density of 1.936 slugs/cu ft.
- b) One of the runs is at a Froude number of 0.4, the hull speed of the ship (g equals 32.17 ft/sec/sec). The measured total model resistance is 13.09 pounds at a model speed of 9.165 ft/sec. Determine the total, frictional, and residual coefficients of resistance for the model if the model Reynolds number equals 1.4×10^7 .
- c) If the frictional-resistance coefficient for the ship is 0.00146 and the correlation allowance is 0.0004, determine the total resistance and the EHP for the FFG7 at a Froude number of 0.4.

11-10. It is desired to run a model test of a DD 993-class destroyer in the 120-ft towing tank. The ship is 530 feet long on the waterline, displaces 7,800 tons in salt water with a density of 1.991 slugs/cu ft, and has a wetted surface area of 33,150 square feet.

The model is 6.625 feet long on the waterline and is tested in fresh water with a density of 1.936 slugs/cu ft and a viscosity of 1.092×10^{-5} ft²/sec.

- a) Determine the scale ratio, the wetted surface area of the model, and the displacement of the model.
- b) One of the runs simulates the full-scale ship operating at 20 knots. The measured

total model resistance is 0.353 pound. Determine the model velocity and the total-resistance coefficient for the model.

- c) The correlation allowance for the ship is 0.0004. Determine the residual-, fric-

tional-, and total-resistance coefficients for the ship.

- d) Determine the total resistance and the EHP for the ship at 20 knots in standard seawater.

Propellers and Propulsion Systems

12-1 Introduction

The first efforts to use mechanical forces to propel ships were made in a tentative way, with indifferent success, probably more often and earlier than is generally realized. Continual frustration with the unreliable and inadequate force of the wind on sails forced men to search for other means to move ships in the water from the earliest days of history. Aside from the simple paddle or manned oars, there is evidence that pre-Christian Romans used paddle-wheel-propelled boats (whose source of power was oxen) to transport soldiers to Sicily. Paddle wheels were apparently used by the Orientals as early as the seventh century, and of course, Leonardo da Vinci designed many mechanical devices for propelling ships.

True mechanical propulsion, however, deriving its power from the energy conversion in a steam engine, came very much later and after many frustrating failures. It is difficult to say where and when such propulsion was first successful, but it is recorded that in 1783, in Lyons, France, a barge-like boat 148 feet in length, equipped with a horizontal double-acting steam cylinder that drove side paddle wheels, was able to move against the current of the Rhone River. This vessel was aptly named the *Pyroscaphe*. Her inventor-designer, Claude de Jouffroy D'Abbans, is generally accepted as the pioneer in the application of steam-powered propulsion to ships. In America, John Fitch of Philadelphia built and experimented successfully with steam-powered vessels as early as 1785 and can be credited with the building of the first commercial steamboat. In 1790 his steamboat, the *Experiment*, began carrying passengers between Philadelphia and Trenton on a regular schedule. His vessel was not propelled by paddle wheels, however. Her 18-in single-cylinder engine and fire tube boiler powered three "duck leg" paddles at the stern, which moved the 60-ft boat at the respectable

speed of 8 knots. James Rumsey of Berkley Springs, Virginia, produced a steam-powered boat in 1787 that was propelled by a water jet. While this boat was intended for ferry service on the Potomac, she was laid up after a successful public demonstration, where she attained a speed of approximately 4.5 knots.

It is interesting to note that in these early efforts, the propelling devices were all dissimilar. The paddle wheel, the mechanical oar, and the water jet all achieved some degree of success a full half-century before the *Archimedes'* screw propeller was successfully adapted by John Ericsson for the U.S. Navy and Francis Petit Smith for the Royal British Navy. Both men took out patents for screw propulsion in 1836 and proceeded to demonstrate the advantages of the screw propeller over the paddle wheel. The screw propeller is less affected by changes in draft and by severe rolling, requires less effective beam, is well protected from damage, and can operate at higher speeds, using more compact machinery. In 1845 the British Admiralty sponsored a famous "tug-of-war" between the steam warship *Rattler*, which was screw propelled, and her slightly smaller and less powerful sister ship, the *Alecto*, which was paddle powered. Although the *Alecto* was given the advantage of moving off first and towing the *Rattler* astern at 2 knots, the *Rattler* arrested her sternway in 5 minutes and proceeded to tow the thrashing *Alecto* backwards at 2.8 knots (Brown 1977). Although it has been shown that the *Alecto* could have won some of the races held as part of the trials if she had possessed equal power, the screw-propulsion detractors had been silenced and warship propulsion was permanently headed toward the use of screw propellers.

There are now many types of ship-propulsion devices. Unusual and inefficient devices were invented, tried, and discarded. The paddle wheel, which was used successfully on the *Great Western* and many other

steamers during the middle of the nineteenth century, has subsided into near obscurity as an open-ocean propulsion system. While it still may exist in some remotely located riverboats and special-purpose craft, it has certainly passed its days of significance. Fitch's walking oars were never used again. The water jet has enjoyed a recent successful rebirth and is used with increasing promise (it will be discussed subsequently in this chapter). The marine propeller with its many variations is the prime propulsive device of modern ships.

12-2 Propelling Devices

Of the successful types of propulsive devices presently in use, the following may be grouped in four distinct categories:

1. Screw propellers
 - a) fixed-pitch propellers
 - b) adjustable-pitch propellers
 - c) controllable-pitch propellers
 - d) shrouded screws working in tunnels or sleeves (ducted propellers)
 - e) contra-rotating propellers
2. Paddle wheels, either side or stern mounted with fixed or feathering blades
3. Jet propellers
 - a) water jet through submerged nozzle
 - b) water jet through surface nozzle
4. Vertical-axis (cycloidal) propellers
 - a) Kirsten-Boeing propeller
 - b) Voith-Schneider propeller

These above types will be individually discussed in the subsequent sections.

SCREW PROPELLERS

Because the most widely used propeller is the screw propeller (referred to henceforth, in keeping with common practice, as a propeller), it will be discussed in the greatest detail. Some general propulsive theory that is applicable to other types of propellers as well will also be covered.

It will be useful to consider first the propeller itself in general terms, along with some associated terms and definitions. A propeller has at least two blades projecting from a hub that is keyed to and driven by the propeller shaft. There are three general types of marine propellers in use today. *Fixed-pitch propellers* have blades that are either an integral part of the hub or are bolted to the hub. In this type of propeller, the position of the blades relative to the hub cannot be altered, with the exception of minor adjustments that may be made during the assembly of some of the bolted-blade types. *Adjustable-pitch propellers* have blades that can be adjusted to different pitch settings when the propeller is stopped. *Controllable-pitch propellers* are provided with a mechanism for altering the position of the blades rel-

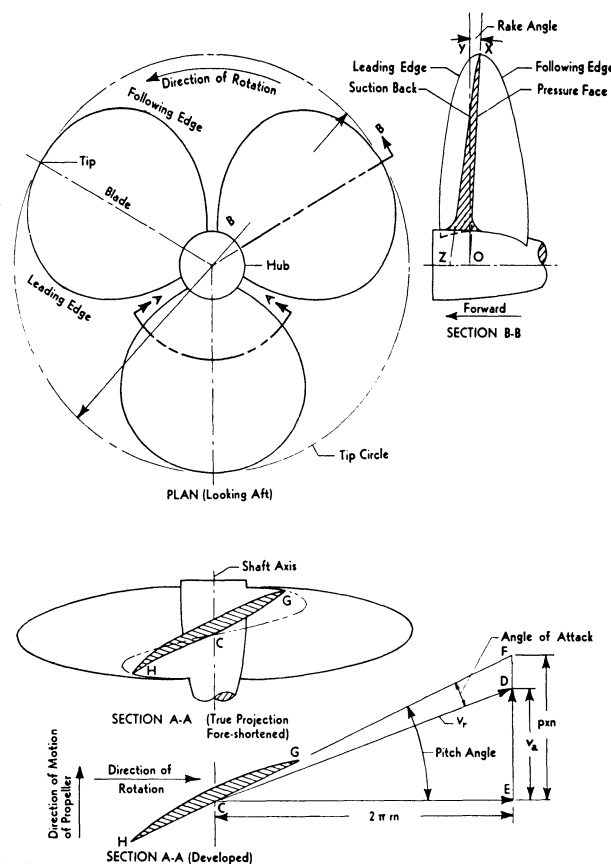


Figure 12-1. Propeller definition diagram (three-bladed, right-hand, constant-pitch propeller)

ative to the hub at any time. In the following discussion, refer to figure 12-1, which shows a three-blade propeller of constant-pitch.

A *right-handed propeller* is one that rotates clockwise when viewed from astern (counterclockwise when viewed looking aft as in figure 12-1), while driving the ship ahead. A *left-handed propeller* rotates counterclockwise when viewed from astern, while driving the ship ahead.

The *pressure face* of a blade is the after side when going ahead.

The *suction back* of a blade is the surface opposite the face.

The *tip* of a blade is the point farthest from the axis.

The *leading edge* of a blade is the edge that cuts the water first when going ahead.

The *following* or *trailing edge* is opposite the leading edge.

The *diameter* is twice the perpendicular distance from the axis to the blade tip, or is the diameter described by the blade tips.

A *helical surface* is a surface generated by a line (the *generatrix*) at an angle with an axis through one of its extremities that revolves about this axis at a constant angular rate and advances along the axis at a constant linear speed. In its simplest form, the pressure face is a portion of a helical surface with the axis along the

propeller shaft. Any surface of the thread of a machine screw is a *helicoid*.

The *pitch* of any point on a blade is the distance moved parallel to the shaft axis by the generatrix of the helicoidal surface through the point in 360° of rotation. The pitch of point C in figure 12-1 is the distance FE for one revolution. When the pressure face is a helicoidal surface, each point on the pressure face has the same pitch, and the propeller is said to be *constant* or *of uniform pitch*. It may be seen in figure 12-2 that each point on the developed blade section has the same pitch as point C. Because it is a constant-pitch propeller, every point on other blade sections will have the same pitch as point C. If the pitch increases from the leading to the following edge, the pitch is *axially increasing*. If the pitch increases from hub to tip, the pitch is *radially increasing*. When the pitch of the blade varies from point to point, the pressure face is not a helicoidal surface.

The theory of propeller design with *variable pitch* over the blade is a particular and specialized adaptation. To summarize the purpose of such design it is sufficient here to say that (1) when the pitch varies between leading and following edges, the propeller will be adaptable to a greater range of ship speeds, and (2) when the pitch varies between root and tip, the propeller will take advantage of the variation in velocities of the wake current around the propeller. The former modification extends the range of efficiency and the latter increases the peak efficiency.

SLIP AND THRUST

There are many mechanical devices that employ the principle of advancing an object by means of a rotating helical screw; some common examples are wood screws, worm gears, and many types of positive-displacement pumps and compressors. In almost all of the above examples, the primary energy losses are frictional, and the velocity of the object or fluid moved is equal to the axial velocity of advance of the helix as determined by its pitch. However, in the case of a ship and its propeller operating in open unobstructed water, the axial advance per unit time of the propeller's helix is not in any case equal to the velocity of the ship. This difference in velocities, called *slip*, is the result of the fluid shear caused by the acceleration sternward of the mass of water ahead of the propeller. Actually, the slip referred to above is apparent slip, and is more precisely defined as a ratio; thus,

$$s_A = \frac{(Pn) - v_s}{Pn} \quad (12-1)$$

where s_A = apparent slip ratio

P = pitch in feet

n = rps

v_s = ship's absolute speed in ft/sec.

Assumed values of slip were utilized in predicting the speed of the early screw-propelled steamship, such as the *Great Britain* mentioned in the previous chapter.

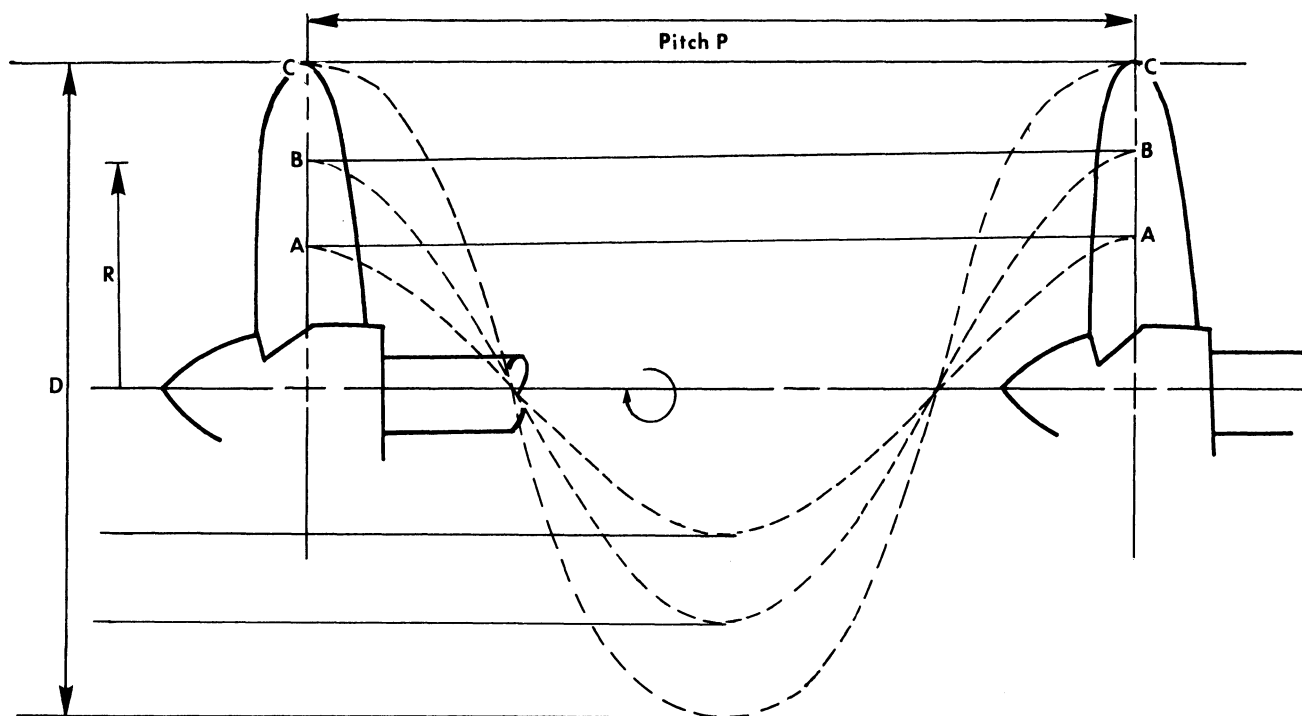


Figure 12-2. Uniform- (constant-)pitch propeller operating at no-slip for one revolution

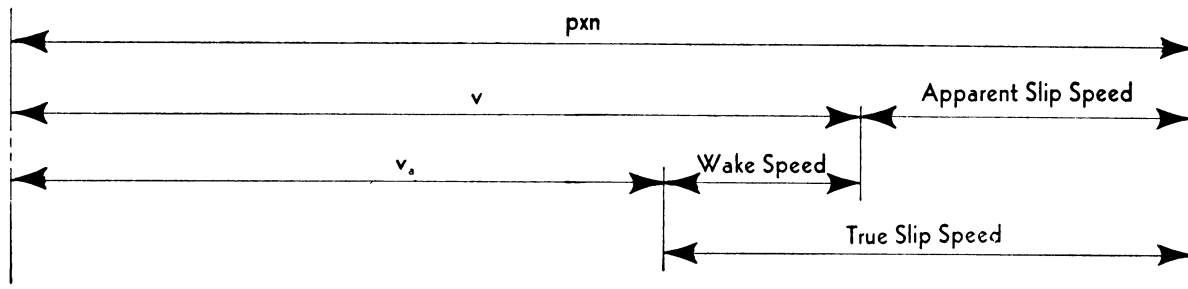


Figure 12-3. Relation of speeds in wake of propeller

The above expression is not altogether representative of the actual amount of slip, because the water surrounding the ship, particularly in the vicinity of the propellers and wake, is disturbed and has its own absolute velocity. It is with relation to this water in which the propeller is working that the true slip must be expressed. This is also expressed as a ratio as follows:

$$s_R = \frac{(Pn) - v_A}{Pn} \quad (12-2)$$

where s_R = true slip ratio

v_A = speed of the propeller relative to the disturbed surrounding water.

From the above expressions it can be seen that with the water in the wake traveling in the same direction

as the propeller, v_A will be less than v_s and the true slip ratio will be greater than the apparent slip ratio. The relationships between true slip, apparent slip, wake, and the propeller and ship velocities may be seen in figure 12-3.

The force produced by the ship's propeller that overcomes the resistance of the ship is referred to as the propeller's *thrust*. Without the slip described above, there would be no thrust. Thrust is the result of change in momentum of the fluid from a point ahead of the propeller to a point astern of the propeller. Because the propeller operates in an unbounded volume of working fluid, it is difficult to resolve the thrust equation to terms of specified dimensions, as is done in pumps, turbojet engines, etc. However, the boundaries of the water set in motion by the propeller may be considered limited, as in figure 12-4 by the dotted line. One must

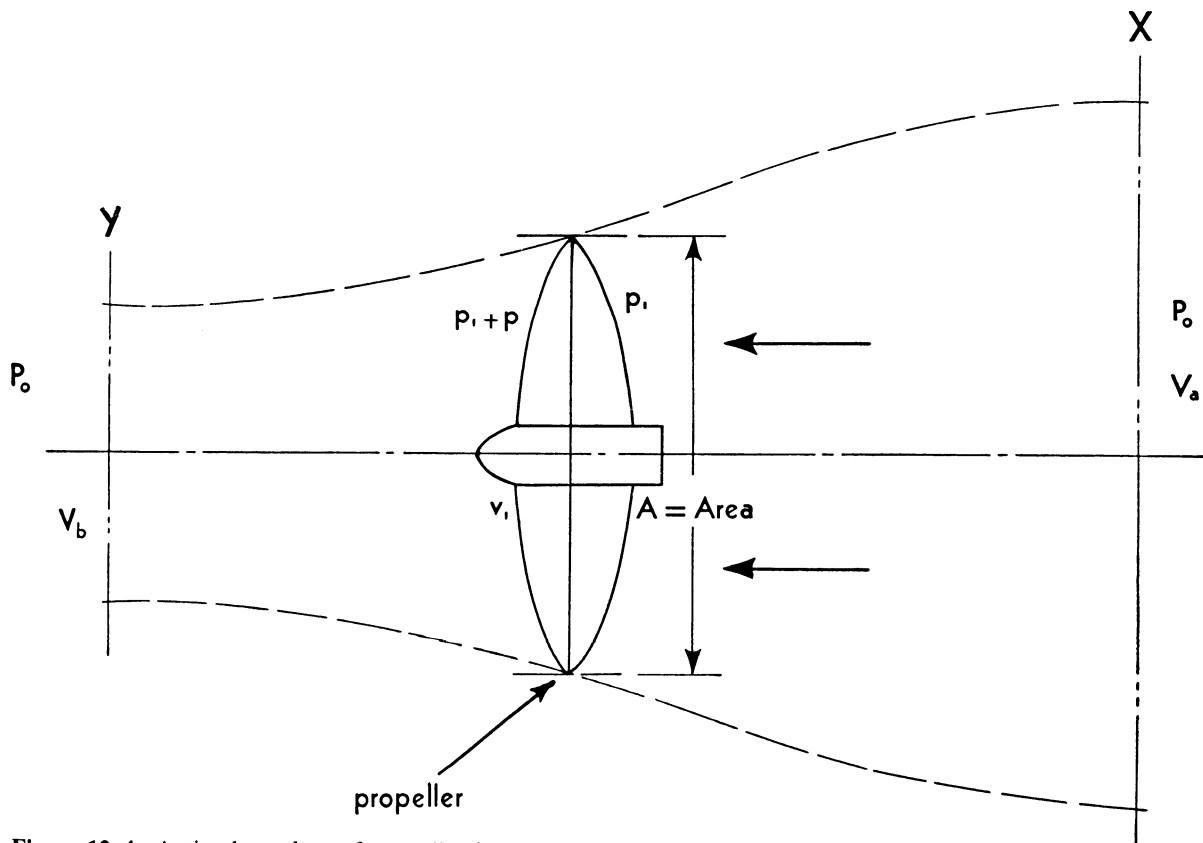


Figure 12-4. Active boundary of propeller jet flow

assume here that the water ahead of section X is at pressure p_0 and at section Y is also returned to p_0 . Assuming that there is no interchange of water between the inside and outside of this imaginary tube, the thrust may be expressed as

$$T = \rho A v_1 (v_b - v_a) \quad (12-3)$$

where T = thrust

ρ = water density (mass)

A = area of the projected propeller disc

v_1 = velocity through the propeller

v_a = velocity ahead of the propeller at X

v_b = velocity behind the propeller.

The expression, as given above, is oversimplified because of the nature of actual propeller installations, and certain assumptions become necessary. However, it does emphasize the basic concept of the conservation of axial-momentum theory, where the reactive thrust on the fluid passing through the propeller is equal to the mass flow rate through the propeller multiplied by the change in velocity; that is,

$$T = \dot{m}(v_b - v_a) \quad (12-4)$$

where $\dot{m} = \rho A v_1$ = mass flow rate.

If the velocity v_a were the same as the pitch times rev/sec, resulting in zero slip, there could be no change in velocity across the propeller; thus, the thrust would be zero. Since energy loss is associated with increasing the change in velocity across the propeller, maximum efficiency for a given thrust is obtained by maximizing the mass flow rate and minimizing the change in ve-

locity. This leads to the use of large, slow-turning propellers on the most efficient merchant ships. Naval hull forms frequently restrict the size of the propeller, and the demand for quick acceleration is incompatible with the high inertia of large propellers and the massive shafting required to handle high torques associated with low revolutions per minute (rpm).

PROPELLER ACTION

Various theories have been advanced to explain actual conditions encountered in propeller operation. The circulation theory gives the best explanation of the phenomenon. This is discussed very briefly in its relation to the forces on a blade section, such as in figure 12-1. The section is advancing along the line CD with a velocity v_r , which is the vector resultant of the speed of rotation $2\pi rn$ and its axial advance v_A . Similar to an airfoil, the blade section is advancing with an angle of attack $\angle FCD$. An unsymmetrical body, such as an airfoil section of a propeller blade, when placed in a parallel fluid flow, will disrupt the symmetry of this flow. The circulation theory indicates that the new unsymmetrical flow can be represented by a counterclockwise circulatory flow superimposed on the parallel water flow past the blade section. The vector sum of the circulatory flow and a parallel flow will produce the high velocity region on the suction back of the blade. The velocity variation will be proportional to the strength of the circulation flow.

By application of Bernoulli's law (equation 11-1), a high velocity region means low pressure and vice versa. It is obvious that the lower velocity on the rear surface or pressure face of the blade increases the pressure of the water on and near that surface in imparting a positive thrust. The higher velocity on the opposite surface

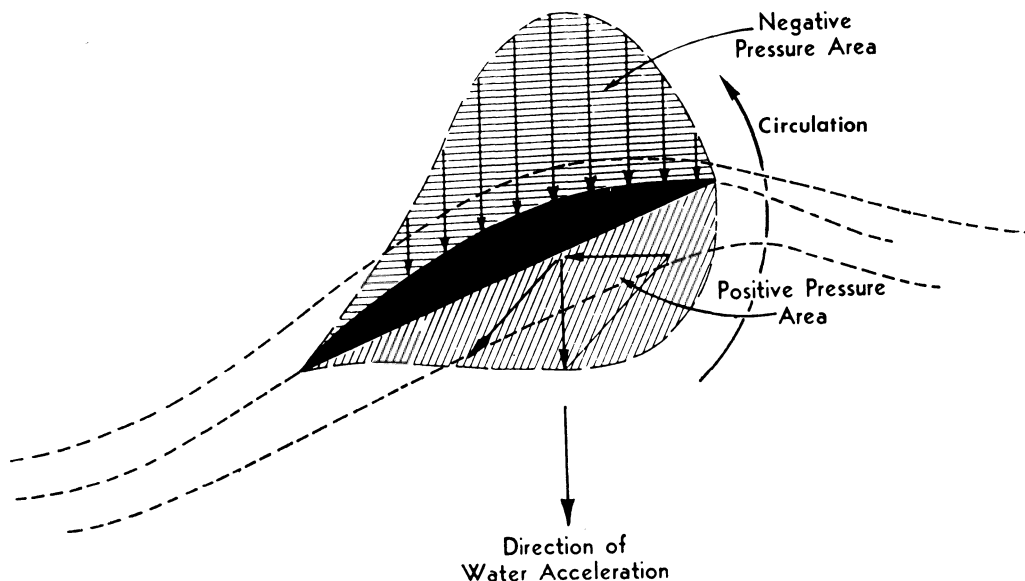


Figure 12-5. Pressure distribution about propeller blade section

of the blade or suction back creates a negative pressure distribution, as indicated in figure 12-5, which may amount to a greater pressure differential than on the pressure face. Together, the total pressure differential on both sides of the blade account for the increase in velocity of the water and for the lift or forward thrust on each blade, from which the overall thrust T is derived. The component at right angles to the shaft is the required force that produces the torque Q .

The pressure distribution from the leading to the following edge of both the pressure face and the suction back is irregular. The pressure reduction on the suction back is greater than the pressure increase on the face, indicating that the greater portion of the propeller thrust is contributed by the suction back of the blades.

CAVITATION

When the minimum value of the absolute pressure on the back is reduced below the vapor pressure of the water, which will occur at relatively high propeller speeds, vapor pockets or cavities are formed that disrupt the flow and reduce the propeller efficiency. This phenom-

enon, known as *cavitation*, generally occurs first in the tip vortex, as illustrated in figure 12-6. When the vapor pockets collapse on the blade surface, erosion of these surfaces results and noise is emitted. Advanced cavitation produces a very slow increase in thrust for increasing shaft horsepower, while speed of rotation increases more rapidly than usual. Fully developed cavitation noise is violent and can be heard easily in the vicinity of the stern.

FULLY CAVITATING PROPELLERS

Although cavitation produces serious blade erosion as well as a breakdown in developed thrust with advancing power, laboratory tests show that a blade with complete cavitation over its back is superior at higher speeds (see figure 12-7). This phenomenon is based on the fact that with complete separation over the back of the blade and no water contact, there can be no further pressure reduction or increasing cavitation effect. On the face of the blade, however, pressure continues to increase with increasing rpm, as does the total thrust. In such applications at high speeds, there are no unsteady and

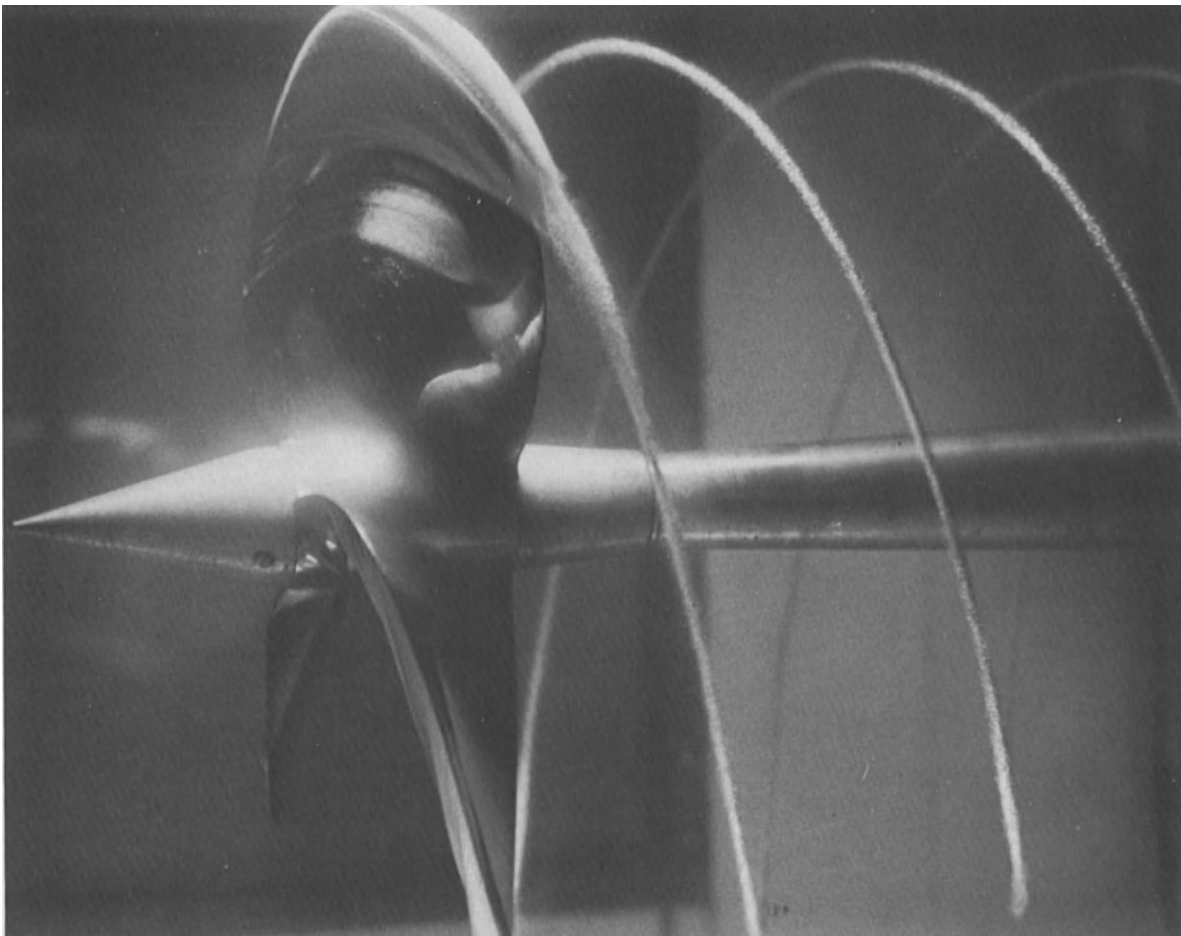


Figure 12-6. Tip cavitation on a common three-bladed propeller in a testing tunnel at the Admiralty Experimental Works, Haslar, England

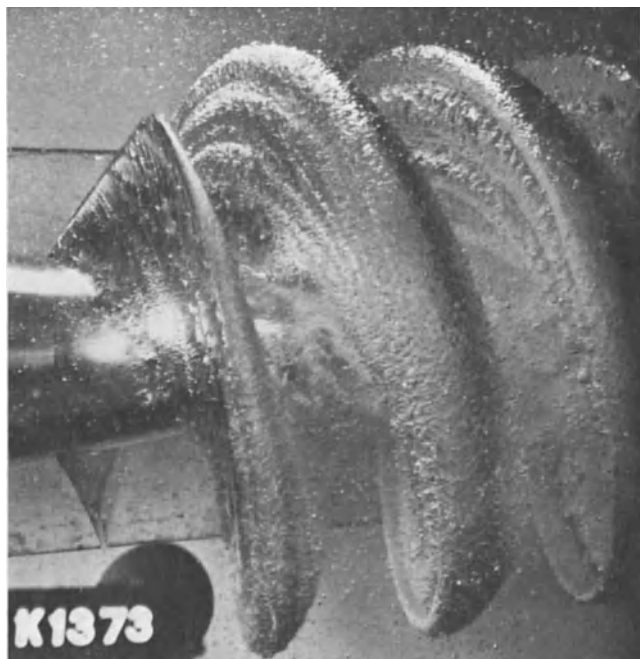


Figure 12-7. Supercavitating propeller model in homogeneous flow. (Courtesy of Swedish State Ship Testing Laboratories [SSPA].)

vibratory forces caused by intermittent cavitation, and there is no erosion of the blades, because the bubbles collapse in the wake. Such propellers are designed with a sharp wedge-shaped blade section to produce clean, complete cavitation. They are less efficient than the conventional subcavitating propellers, but in the high-speed regimes for driving craft in the range of 40 to 80 knots, they offer a practical alternative to water-jet propulsion.

PROPELLER DESIGN

It is possible to design a propeller by means of the circulation theory. At the present writing, this method requires the application of certain corrections to give agreement between calculated and test values. The failure of the theory to substantiate experiments without some corrections indicates that the theory is at present incomplete and that its further development is necessary. The method of propeller design by chart is the most common and produces very satisfactory results. Tests of the model propeller are performed in open, undisturbed water by varying either the propeller rpm or the speed of advance and measuring, in addition to rpm and speed of advance, thrust and torque. These data are converted to a set of nondimensional coefficients (constant regardless of propeller size) and plotted.

Common dimensionless parameters that are used in propeller design and are a means of comparing various

propellers' performances must be familiar to all designers. Some of the most frequently used are

1. Thrust coefficient

$$K_T = \frac{T}{\rho n^2 D^4} \quad (12-5)$$

2. Torque coefficient

$$K_Q = \frac{Q}{\rho n^2 D^5} \quad (12-6)$$

3. Advance coefficient

$$J = \frac{v_A}{nD} \quad (12-7)$$

where T = thrust in pounds

ρ = mass density of water in slugs/ft³

D = propeller diameter in feet

Q = torque in ft-lb

v_A = speed of advance in ft/sec

n = revolutions (rps)

4. Open-water propeller efficiency

$$\eta_0 = \frac{T v_A}{2\pi n Q} = \frac{J}{2\pi} \frac{K_T}{K_Q} \quad (12-8)$$

5. Propeller loading

$$\left(\frac{K_T}{J^4} \right)^{1/4}$$

6. Pitch-diameter ratio

$$\frac{P}{D}$$

The relationship between advance coefficient and slip ratio may be found by combining equations 12-2 and 12-7.

$$J = \frac{v_A}{nD}$$

$$v_A = Pn(1 - s_R)$$

Thus,

$$J = (1 - s_R) \frac{P}{D}$$

The *bollard pull* condition (v_A equal to zero) corresponds to a slip ratio of unity and a J equal to zero. At a slip ratio of zero, J equals the pitch-diameter ratio.

Open-water propeller test data are typically presented as curves of torque and thrust coefficients and propeller efficiency versus advance coefficient J (figure 12-8). The nominal pitch-diameter ratio in this case equals unity, but the thrust does not vanish at the zero slip condition. This indicates an effective pitch greater

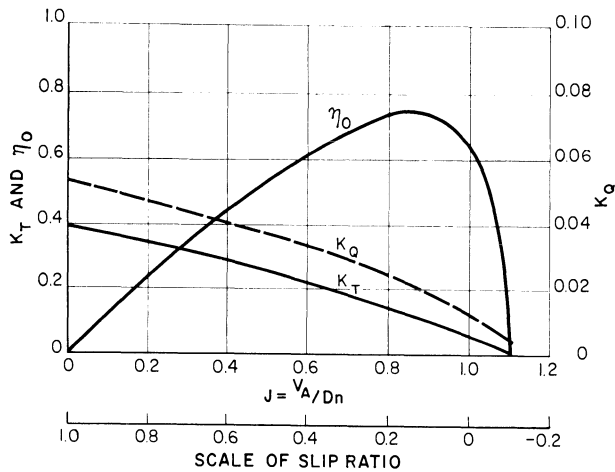


Figure 12-8. Typical curves of thrust, torque, and efficiency for a propeller in open water. (From Comstock 1967.)

than the nominal pitch as a result of cambered (unsymmetrical) blade sections that produce lift at zero angle of attack (see chapter 14). Notice also that the maximum efficiency is achieved at a J of approximately 0.85 or a slip ratio of 0.15. These coefficients are used chiefly in plotting the design criteria for series propeller tests (figure 12-8) and in the construction of the design charts.

A series of design charts have been developed by model tests for groups of propellers with varying pitch ratio. Within each group, other characteristics, such as number of blades, mean-width ratio, blade-thickness fraction, and shape, are varied. The usual design problem is to determine from these charts the propeller that will give the best efficiency by entering with the known information and any restrictions on the design, such as

maximum diameter. This design method restricts the propeller to similarity with a prototype, and will result in the design of the most efficient of the group but not necessarily in the best propeller for the specific requirements. For further information see *Principles of Naval Architecture* (chapter 7).

A single propeller located on the centerline is the most efficient for normal ship forms. The number of propellers is usually determined by limitations on propeller diameter, size of individual propelling plants, internal hull arrangements, the greater immunity to complete breakdown, and ability to maneuver with twin or quadruple screws. The choice of the number of blades ranges between three and nine. The selection of the number of blades may be dictated by hull-vibration considerations, as shown in figure 12-9.

Since nearly all marine propellers operate in the wake of the hull, or of the shaft-supporting struts in the case of multi-screw ships, resonant excitations can be set up as the blade passes through the velocity defect in the wake. When a propeller blade enters the wake, the angle of attack changes suddenly, causing a rapid change in torque and thrust. The excitations can set up hull resonances and severe vibrations that must be avoided by ship designers. (Even-bladed propellers are sometimes avoided on single-screw ships for acoustic reasons, since two blades would pass through the wake simultaneously.) Another method that distributes the effect of the wake more uniformly during a complete revolution of the propeller is to sweep back the leading and trailing edges of the individual blades to produce a *skewed propeller*, as illustrated in figure 12-10.

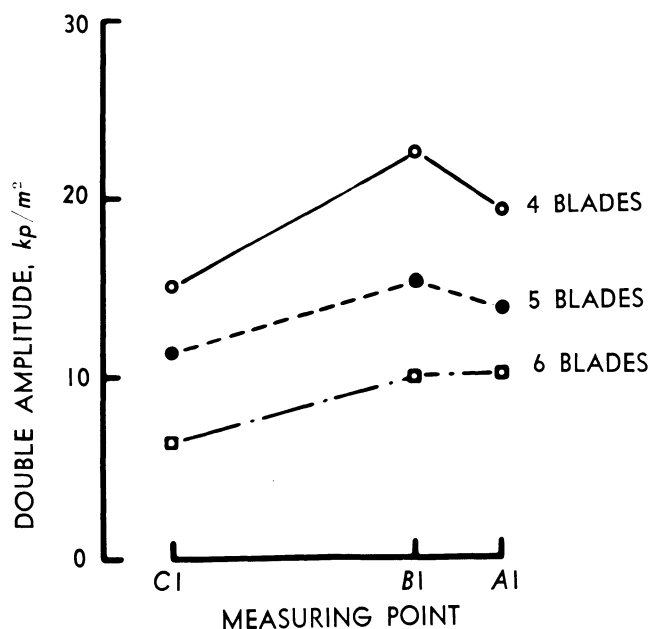
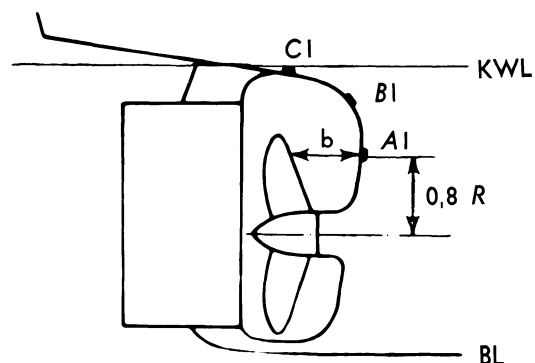


Figure 12-9. Effect of the number of blades on the pressure impulse amplitude (vibration)



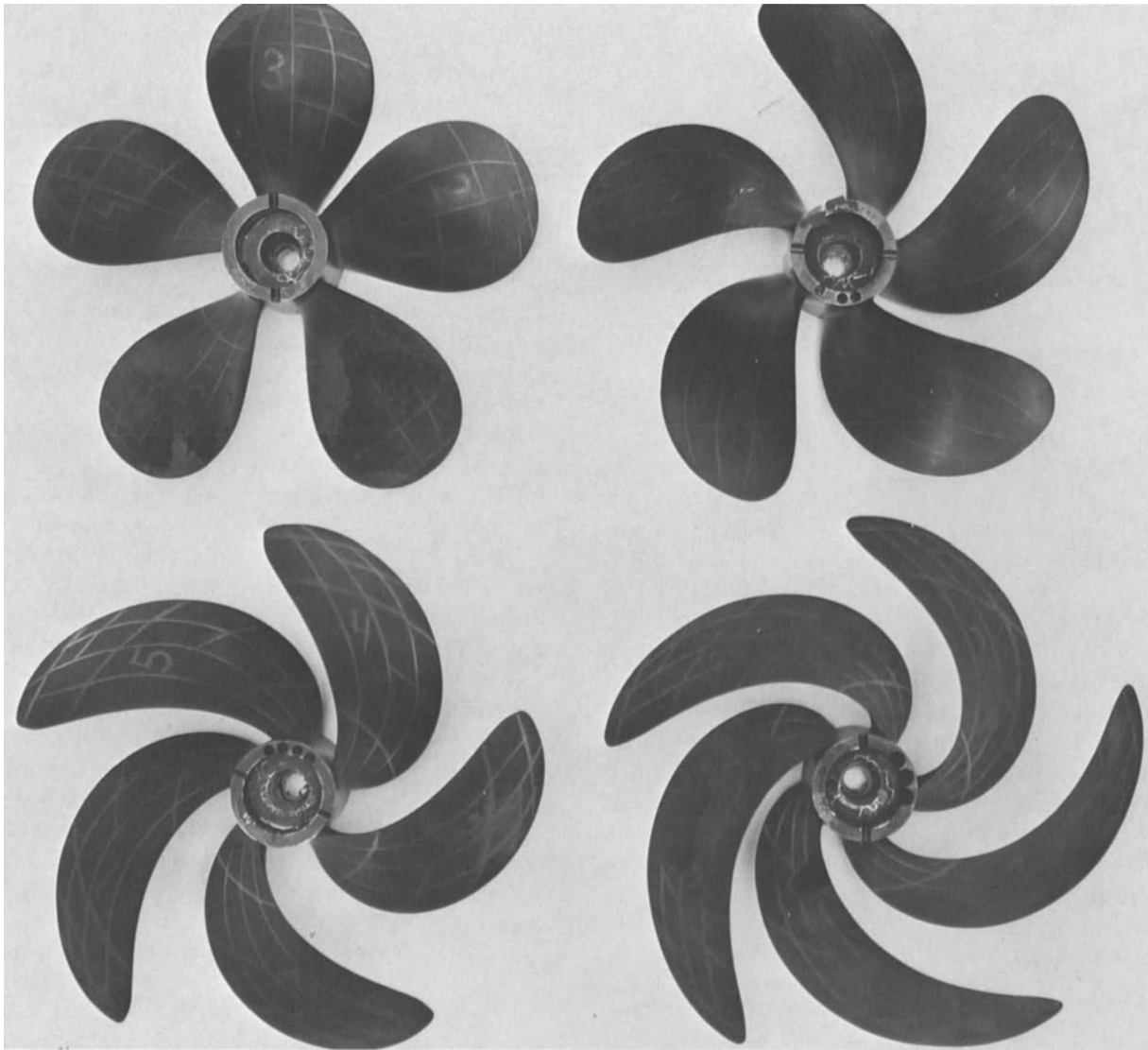


Figure 12-10. Longitudinal view of the skewed-propeller series

Variations from designed dimensions are usually evidenced by abnormal vibration. Imbalance, pitch variations between blades, failure of blades to track (corresponding points on all blades lying in a circle whose plane is normal to the propeller axis), and variations in blade thickness result in increased vibration. Such variations usually decrease propeller efficiency and frequently cause cavitation. If surface erosion is noted when in dry dock, indicating cavitation, the propeller dimensions should be checked.

CONTROLLABLE-PITCH PROPELLERS

In some types of propulsive power plants, such as large marine diesels and gas turbine plants, it is difficult to reverse the direction of the shaft. To do this often requires large and expensive reversing mechanisms and generally involves an appreciable interval of time in any type of plant. In addition, the efficiency characteristics

of the power plants are such that a peak is reached at a given rpm. To assist in overcoming these difficult features, in many large ships there are propellers equipped with blades that rotate from full-ahead pitch to full-astern pitch. The bridge-activated mechanism that accomplishes this consists of servo-controlled hydraulic pistons that transmit motion through linkages or hydraulic lines in a hollow propeller shaft to a geared drive or piston in the hub or the propeller. This mechanism in turn transmits the motion to each blade equally to change its pitch (see figure 12-11).

The controllable-pitch propeller, in addition to full reversing without changing the direction of the shaft rotation, extends the range of propeller efficiency, which in a fixed-pitch propeller has a characteristic peak at a given advance coefficient J . The proper pitch may be adjusted for every engine speed. Such a feature is particularly advantageous for tugboats, landing craft, and

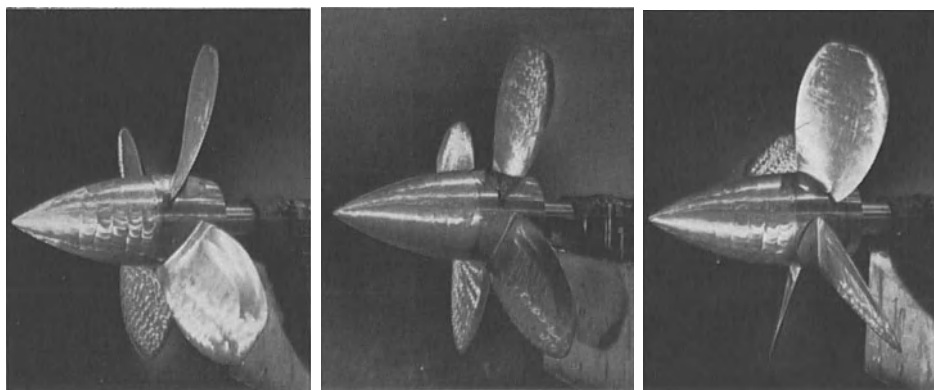


Figure 12-11. Controllable-pitch propeller. Full-ahead position is (left), no-thrust position (middle), and full-reverse position (right). (Courtesy of Morgan Smith Co.)

mine sweepers, where high torque absorption is important for various ranges of ship speed and engine revolutions.

PROPELLER SHROUDING AND CONTRA GUIDES

Because of the interaction between the water in the “jet-stream tube” and the surrounding undisturbed water, it is possible to recover some of this energy along with the tip losses of the propeller by surrounding it with a short cylindrical tunnel or nozzle (see figure 12-12). This shrouding or ducting, as it may be generally termed, directs the stream of incoming water directly into the propeller disc area, effecting generally a greater change in velocity of the water and increasing the propeller efficiency. The shrouding should be provided with the necessary structural dimensions and material to make it a true, streamlined tunnel in the form of a nozzle. (A very well-known device of this sort is the patented Kort nozzle.)

Because of the rotary motion imparted to the water as it passes through the propeller from slightly ahead of it, some of the effective velocity component is lost by virtue of the water’s rotational motion. To assist in straightening out this helical motion, some propeller shaft struts and some types of rudders are shaped to direct this stream out of a rotational path. In the case of some rudders directly in the propeller stream, the upper half is angled several degrees in one direction and the lower half angled the same amount in the opposite direction. Shaft struts nearest the propeller are similarly angled, but only in one direction, since they are effective on only half of the propeller stream. Such devices are sometimes called *contra guides*.

Contra guides increase the efficiency when the ship is going ahead, but they give a corresponding decrease in thrust when going astern and are therefore seldom used in ships where maneuverability is important.

COUNTER ROTATION

In the further pursuit of a means to recover the rotational losses imparted to the stream of water through the propeller and to increase the horsepower available per shaft line, coaxial propellers (two or more propellers on one axis) have been used in a limited way in the past. They are more recently enjoying a new significance for large ships, where increased rpm is impractical but increased blade loading is unavoidable.

It is not possible, of course, to recover all of the rotational energy lost by the first propeller on the shaft. Because of the interference between propellers caused by the aft propeller working in the disturbed wake stream, the propeller efficiency is lower than for a single propeller of the same loading. The propeller circle is reduced in diameter, however, and the overall propulsive efficiency is higher because of the increased hull efficiency and relative rotative efficiency. These generalizations are valid primarily for comparatively low rpm. Counter-rotating propellers are inadvisable for higher-speed rotations because of the effects of cavitations of the forward propeller on the after one (see figure 12-13). In addition, the required gearing and concentric drive shafts make an expensive and complex installation that may be an important consideration in the life-cycle costs.

In designing counter-rotating propellers, the pitch of the after propeller must be greater than that of the forward to be compatible with the increased water velocity entering it. Also, to avoid vibration problems, the number of blades should be different on each of the two propellers, so that no more than two blades are passing each other at any time.

WATER-JET PROPULSION

Water-jet propulsion in its modern sense can be defined (or better described) as the propulsive force provided

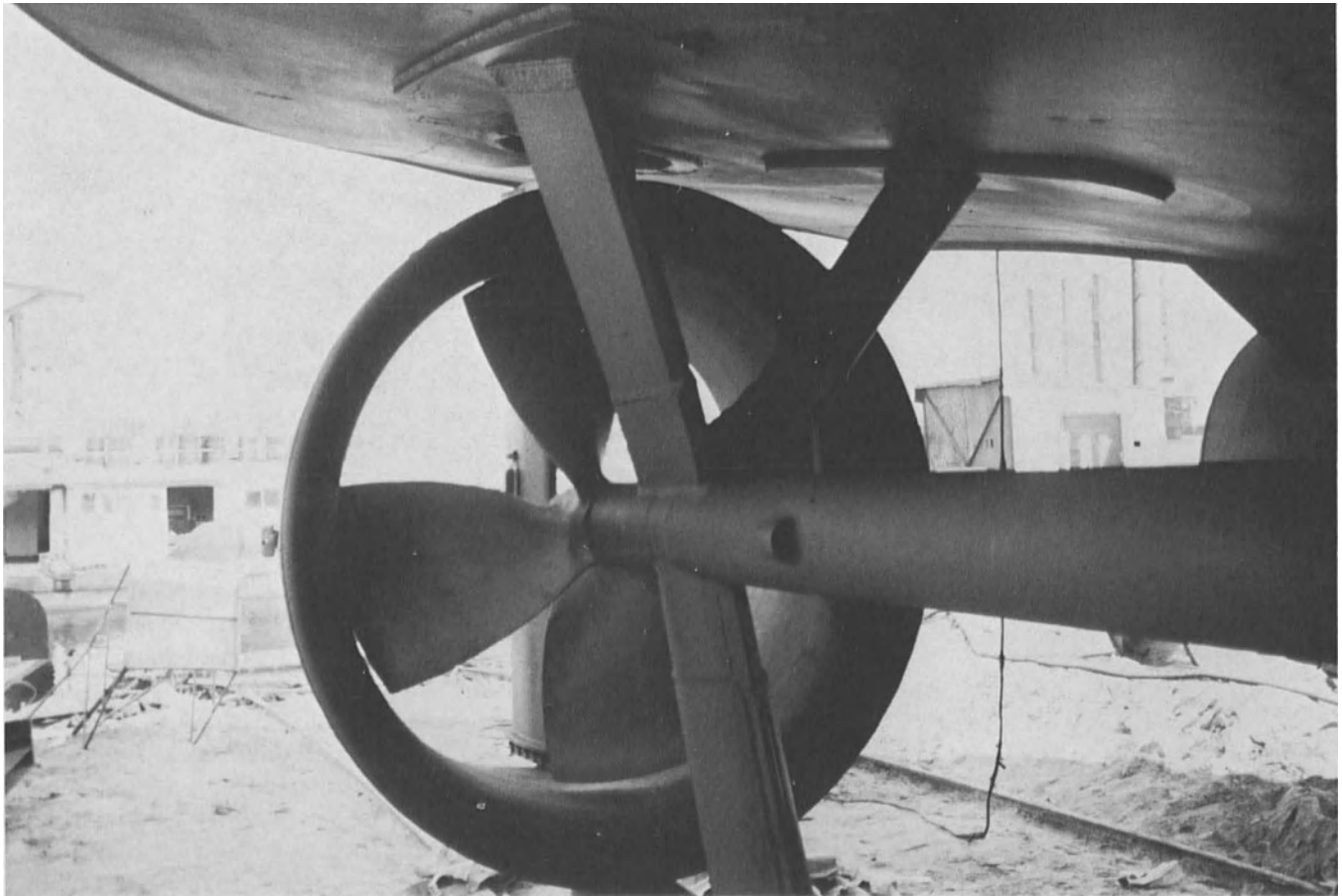


Figure 12-12. Ducted propellor. (Courtesy of Michigan Wheel.)

by the jet-reactive thrust of high velocity water expelled through a nozzle. The theory is similar to the basic momentum theory reviewed in connection with the marine propeller and is an extension of it. The application is most common to the propulsive drive of jet aircraft. Rather than using air as the working substance, however, water is fed into an internal high-powered pump or impeller, which adds energy in the form of high

velocity and expels it aft through the nozzle. It might also be thought of as an internal, ducted propeller.

As an alternative to super-cavitating propellers for high-speed craft and some special-purpose craft, the water jet, driven by gas turbines, has shown considerable promise. It is presently being used in the most recent naval hydrofoil craft (figure 12-14).

In the lower speed ranges, water-jet propulsion is

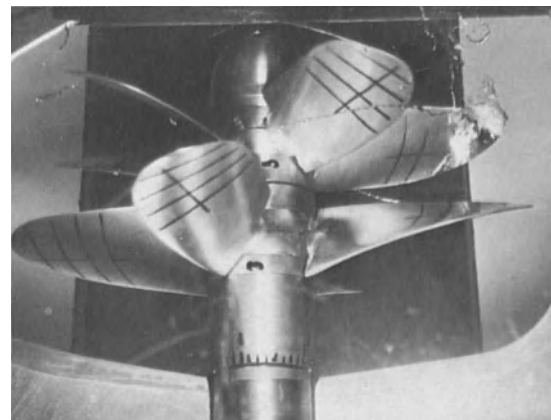
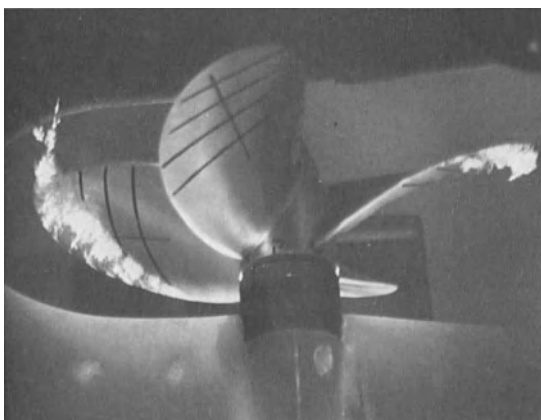


Figure 12-13. Conventional (left) and contra-rotating (right) propeller models in test tunnel (note incipient cavitation). (Courtesy of SSPA.)



Figure 12-14. The USS *Taurus*, a patrol hydrofoil missile ship (PHM), makes a foil-borne high-speed turn in Puget Sound off downtown Seattle. (Courtesy of Boeing Marine Systems.)

inefficient. Thrust depends upon a change of momentum, and the jet develops less thrust at low and intermediate speeds than do conventional propellers because of the inherent difficulty it has handling large mass flow rates. However, as speed increases, the characteristics of jet propulsion show an increase in propulsive efficiency, whereas the opposite is true for conventional propellers. Consequently, water-jet propulsion is of advantage in these higher-speed regimes, not only for hydrofoil craft, but also possibly for other high-speed applications, where it might be combined with a practical means of boundary-layer ingestion.

A schematic diagram showing a typical water-jet installation is given in figure 12-15.

While the water jet in the past was confined to lower speeds and powers and hence deemed a poor means of propulsion, its future is now brighter, since it can be combined with higher-powered, lightweight gas turbines.

A further attraction of water-jet propulsion is the almost total lack of any projecting appendage below the hull. This advantage has been profitably used in many small craft that must operate in shallow, restricted waters.

VERTICAL-AXIS CYCLOIDAL PROPELLERS

A later development in the field of marine propellers is a device generally referred to as a *vertical-axis propeller*. There are two types of vertical-axis propellers,

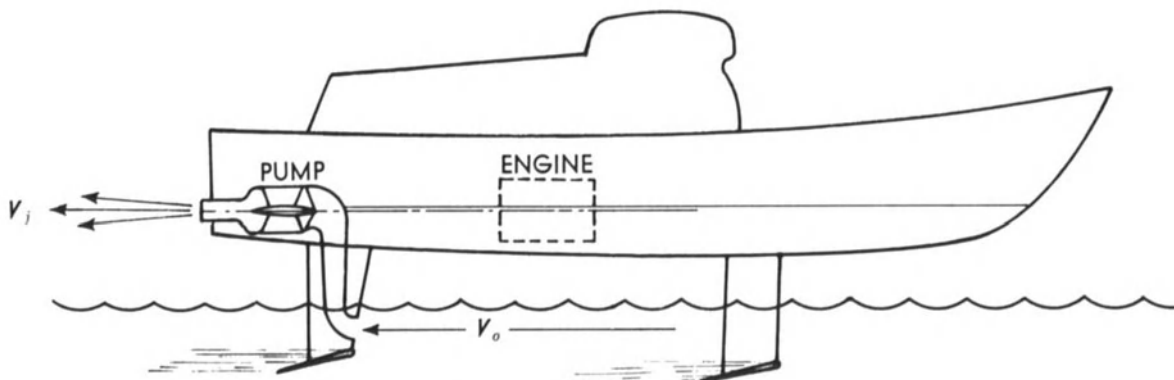


Figure 12-15. Definition sketch of water-jet propulsion system with hydrofoil craft



Figure 12-16. Vertical axis (Voith-Schneider) propeller installation on a US Army towing vessel. (Courtesy of the Dravo Corporation.)

which differ in details of operation but are based on the same basic theory. These are known by the names of their inventors: the Kirsten-Boeing propeller and the Voith-Schneider propeller. Both of these propellers consist of four or more blades projecting from a circular disc, whose axis is vertical and generally flush with the bottom of the hull. This disc is geared to the propeller drive shaft, and as it rotates, the blades are capable, by means of cam action, of maintaining a positive angle of attack when they are in a fore-and-aft line with the axis, and a zero angle or flat surface to the stream when in the athwartships position. The position of the cam with respect to the disc may also be varied so that reverse or side thrust may be produced. This type of device provides generally a very versatile propeller, with both turning and reverse action that eliminates the need for both a rudder and reversing mechanisms at the engine. Here again, the efficiency of this device is not as high as that of the screw propeller, but it has maneuverability characteristics that are superior to those of any other type of propeller. It has been used with considerable success in small harbor craft, patrol boats, and towboats, where the maneuverability feature is highly desirable. Figure 12-16 shows a typical installation of a vertical-axis propeller used in an army towboat, where maneuverability in both pushing and towing is of maximum importance.

12-3 Propulsion Efficiency

In discussing the power transmission from the ship's power plant to its terminus, it is useful to include certain traditional expressions and relationships in order to assign the propulsive losses or efficiencies or both. The following definitions must therefore be used (see figure 12-17):

1. *Brake horsepower (BHP)* is the power at the engine.

2. *Shaft horsepower (SHP)* is the power measured in the shafting within the ship by a torsion meter as close to the propeller or stern tube as possible. It is equal to the PHP plus the bearing losses between the propeller and the torsion meter.

$$\text{SHP} = \frac{2\pi n Q_s}{550} \quad (12-9)$$

where Q_s = torque measured on the shafting just inside the stern tube.

3. *Propeller horsepower (PHP)* is the power delivered to the propeller. It is equal to the effective horsepower, plus the power losses in the propeller and the losses in the interaction between the propeller and the ship.

$$\text{PHP} = \frac{2\pi n Q_D}{550} \quad (12-10)$$

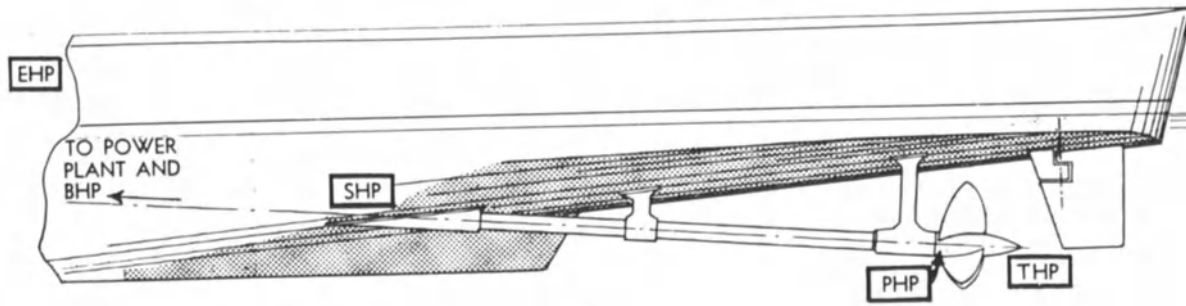


Figure 12-17. Power relationship along the propeller shaft

where Q_D = torque delivered to the propeller.

4. *Thrust horsepower (THP)* is the power delivered by the propeller.

$$\text{THP} = \frac{T v_A}{550} \quad (12-11)$$

5. *Effective horsepower (EHP)* is the power required to tow a ship without its propulsive device.

$$\text{EHP} = \frac{R_T v_S}{550} \quad (12-12)$$

6. *Shafting efficiency* is the only efficiency based on mechanical losses.

$$\eta_s = \frac{\text{PHP}}{\text{SHP}} = \frac{Q_D}{Q_s} \quad (12-13)$$

INTERACTION BETWEEN HULL AND PROPELLER

Up to this point, the resistance of the ship and the action of the propeller have been considered separately. It is now necessary to consider the interaction between the hull and propeller when the ship is self-propelled. In the discussion of resistance, a wake or wake current was mentioned. This wake, which is the motion of the water immediately surrounding the ship relative to undisturbed water, is considered positive when moving in the same direction as the ship. It is the algebraic sum of three components: the frictional wake, the streamline wake, and the wave wake, associated with the orbital velocities in the transverse wave generated by the ship. It is equal to v_S minus v_A . The wake speed is customarily defined as a fraction of the ship's speed v_S . Thus,

$$w = \frac{v_S + v_A}{v_S}$$

or

$$v_A = (1 - w)v_S \quad (12-14)$$

where w = wake fraction.

The propeller of an actual ship does not work in undisturbed water as in an open-water propeller model

test, but in water disturbed by the wake current it experiences a change in the relation between thrust and torque from the open-water condition (equation 12-8). The ratio between thrust and torque, as measured in self-propelled and open-water tests, is called the *relative rotative efficiency*.

$$\eta_R = \frac{T}{Q_D} \frac{Q_0}{T_0} = \frac{\eta_B}{\eta_0} \quad (12-15)$$

Values of η_R range from 95 to slightly over 100 percent. The efficiency of the propeller operating behind the hull can be expressed in the equation

$$\eta_B = \frac{\text{THP}}{\text{PHP}} = \frac{T v_A}{2\pi n Q_D} = \eta_0 \eta_R \quad (12-16)$$

The propeller acts on the ship by increasing the water velocity near the stern and creates an augmentation of resistance. For a given speed, the propeller thrust T is greater than the ship's total resistance R_T . The quantity, T minus R_T , is called the *thrust deduction* and is normally expressed as a fraction of the thrust.

$$t = \frac{T - R_T}{T}$$

or

$$R_T = (1 - t)T \quad (12-17)$$

where t = thrust-deduction coefficient
 $1 - t$ = thrust-deduction factor.

The net effect of the wake and thrust deduction on the hull is called the *hull efficiency*.

$$\eta_H = \frac{1 - t}{1 - w} = \frac{R_T v_S}{T v_A} = \frac{\text{EHP}}{\text{THP}} \quad (12-18)$$

In general, values of t and w are of the same magnitude. Hence, the hull efficiency is usually in the vicinity of 100 percent.

The ratio between the effective horsepower and the

power delivered to the propeller is called the *propulsive* or *quasi-propulsive efficiency*.

$$\eta_D = \frac{\text{EHP}}{\text{PHP}} = \eta_H \eta_0 \eta_R \quad (12-19)$$

And finally, the *propulsive coefficient (P.C.)* is the ratio between effective horsepower and shaft horsepower.

$$\begin{aligned} P.C. &= \frac{\text{EHP}}{\text{SHP}} = \eta_D \eta_s \\ &= \eta_H \eta_0 \eta_R \eta_s \end{aligned} \quad (12-20)$$

From the foregoing discussion in this chapter and the preceding chapter on resistance, one can see that proceeding from the effective horsepower, determined by model test, to the brake horsepower at the engine is a complex task. There are many losses through this energy transmission, including mechanical, thermal, and fluid losses, many of which evade precise analysis.

The most troublesome power losses to account for are those fluid losses through the propeller. The performance characteristics of a propeller in a towing tank or a cavitation tunnel, where it can be tested and analyzed, cannot very closely duplicate conditions when installed in its operating position on the ship. The thrust deduction and the wake phenomenon often have the opposite effect and tend to cancel each other out. They are also erratic at varying speeds. There is no completely satisfactory method of analysis for the quantitative measure of these factors. One approach for the designer is to use the best figures available from full-scale power trials of similar ships, drawing from experience where there are design dissimilarities. Model self-propulsion tests, when the facilities are available, provide the required factors.

Another difficult determination is the power delivered at the propeller. Because at this point the propeller shaft is outside of the ship, it is not practical to make torsion measurements under operating conditions. However, by adding the best estimates of the fluid losses through the propeller to the effective horsepower, calculating the strut bearing or stern tube-bearing losses, and proceeding through the internal shaft bearings and gear losses, the designer can arrive at the engine requirements with only a slight diminishing of his confidence. But with today's computer memory banks and rapid data processing, there is substantially less opportunity for error. The great mass of accumulated data on performance characteristics, which contains accurate trends for the power losses for a multiplicity of designs, is now providing opportunities for far more thorough analyses than ever before.

ESTIMATION OF SHIP REQUIREMENTS FROM MODEL SELF-PROPULSION TESTS

The details of self-propelled model testing for the estimation of SHP are beyond the scope of this text. Open-water propeller characteristics are usually determined in a towing tank, where cavitation effects are not scaled (except in the large vacuum tank in the Netherlands). Propeller characteristics are definitely Reynolds-number dependent, so the model propeller should be as large as possible within the capabilities of the tank and the funds available for the test. The wake of the model and full-scale ship in which the propeller operates can be quite different, so that care must be taken in scaling up the wake fraction and thrust deduction and in interpreting the efficiency of the propeller operating behind the hull. There is considerable activity within the ITTC to develop empirical scaling methods in which all effects are accounted for, and reasonable confidence can be assigned to the powering estimates.

12-4 Operational Factors Relating to Power

There are a number of other factors encountered in operation that must be allowed for in power determination and selection. Customarily, speed-power tests in model basins are made in flat, calm, fresh water. The allowance for sea water is made in the calculations to full scale as noted. The effects of shallow water, rough water, and wind resistance were briefly noted in chapter 11; however, in order to emphasize the order of magnitude to be expected and any possible allowances, they will be mentioned in the context of quantitative allowances here. Other factors, such as displacement changes and trim changes, must be considered in the design process, where the mission and employment of the vessel indicate the importance of these variables.

DISPLACEMENT

Changes in displacement will alter the form and amount of the wetted surface from the still-water condition. Reduction in displacement almost certainly will decrease resistance. Very light displacements may require an accompanying trim by the stern to avoid a serious decrease in propulsive efficiency, resulting from the propeller tip circle being so close to the surface that the blades draw air in smooth water, and from the propeller breaking the surface and racing in a rough sea.

TRIM

A change in the still-water trim will alter the underwater form at the extremities of the ship. The effect on resistance depends on the hull form and on the speed. Trim by the stern may cause additional eddy resistance at low speeds, where wave-making resistance is a minor item. At higher speeds, where wave-making resistance

is predominant, trim by the stern may result in less resistance or more resistance, depending on the hull form. Trim by the stern has some advantages in that it generally improves directional stability, and in rough weather it is desirable to secure greater immersion for propellers and greater freeboard forward.

CHANGE OF TRIM AND SINKAGE

When a ship is under way in calm water, there is a change of level of the bow and stern in response to the elevation and depression of the water level about the hull, caused by the bow and stern wave systems. Up to moderate speeds, the ship usually experiences a body sinkage or squat without any appreciable change in trim. At higher speeds, the sinkage at the bow ceases, and the bow begins to rise while the stern continues to settle. Squatting and changes in trim are symptoms of increased wave-making resistance.

FOULING

In design work, the wetted surface of the ship is considered to be perfectly clean. The effect of fouling is to increase the roughness of the wetted surface and, consequently, the frictional resistance. The extent of fouling may be such that it could increase the clean-bottom resistance by 50 percent. The rate of fouling by sea growth on a ship's bottom is a variable function, depending upon local conditions, water temperatures, salinity, and the amount of time the ship remains in port. It is difficult to predict or allow for. However, with the types of antifouling paint currently in use on steel-hull U.S. Navy vessels, fouling has been reduced so greatly that ships are normally docked for cleaning and painting only during routine overhaul (normally every eighteen months) rather than being docked each nine months.

WIND RESISTANCE

The discussion of air resistance in chapter 11 considered only the relative wind due to the ship's motion. High head winds will cause appreciable increases over the still-air resistance, mainly because of the eddy-forming features of superstructures of conventional design. A head wind having a velocity in the range of one to two times the ship's speed will result in a wind resistance of about 10 to 20 percent of the ship's water resistance. Streamlining the superstructure can undoubtedly reduce the wind resistance caused by winds from dead ahead. However, unlike an airplane, which advances directly into the air stream, the wind more often strikes the superstructure from an angle than from dead ahead. It is extremely difficult to provide effective superstructure shapes for these variable conditions. The wind resistance may reach a maximum value of about 130 percent of the dead ahead wind resistance when the relative wind is about 30° on the bow. Wind may cause a further

increase in resistance if it strikes at an angle requiring a permanent rudder angle to maintain the ship on the desired course.

ROUGH WEATHER

In rough weather the ship's resistance will be increased. The amount of increased resistance is extremely difficult to evaluate, because it will vary with the relative size, speed, and course of the ship and the waves. Model tests are frequently made in towing tanks equipped with wave-making devices, and such tests for powering are becoming increasingly common, to the point that they, together with the tests for various displacements and trim conditions, are considered a part of the total spectrum of speed-power characteristics. In head sea conditions, there will be some critical speed or rate of encounter of a wave system of a particular wave length to ship length ratio $\left(\lambda \text{ equals } \frac{Lw}{Lpp}\right)$, usually where λ is approximately 1.0 to 1.25, that will produce a maximum pitching motion. It is in this circumstance that powering conditions are most disadvantageous, not only because of increased resistance but also because of the reduced propulsive efficiency.

SHALLOW WATER

In shallow water, the restricted passage for water flow around the hull brings about greater water velocities, greater pressure differences, and hence waves of greater height. This normally produces an increase in wave-making resistance. However, as the speed of high-powered ships in shallow water is increased, a critical speed is reached, depending on the ship's length and speed and the depth of the water, where the difference between shallow-water and deep-water resistance becomes a maximum. At speeds higher than the critical speed, the increase in shallow-water resistance diminishes rapidly to zero, and the ship may, at extreme speeds, actually encounter less resistance in shallow water than in deep water.

It is difficult to say at just what depth the effect of increased resistance due to shallow water begins to become effective. Because of the variables, such as speed and draft, which are both functions of this increased resistance, it must necessarily be a dependent relationship. However, if the speeds are moderate (below the critical range mentioned above), it is possible to state an effective depth-speed relationship as

$$V = 0.4 \sqrt{gh}$$

where V = speed in knots
 h = depth of the water in feet.

The above equation indicates depths of water that are not ordinarily thought of as shallow water. However, based on model tests, the effect of increased resistance can be detected as beginning at these drafts. For power and standardization trials, a location is usually selected where the depth of water is greater than that given by the equation above.

Sinkage and changes in trim are greater for any given speed in shallow water than in deep water. To avoid touching bottom in channels where the depth of water affords only a minimum clearance, ships should proceed at very low speeds.

POWER LOSS

As must be evident from the foregoing discussion, the means for determining the power requirements during the design process are imperfect. The difference between the actual power that the ship expends and that which was predicted in the design lies somewhere in the so-called correlation allowance or roughness coefficient (C_A). In terms of percentage error this difference may vary from a very negligible fraction up to 4 or 5 percent. Normally where the hull form is not unusual and where the model testing procedures are thorough, an error of no more than one percent can be expected.

12-5 Resistance and Powering Characteristics of Submersibles

As pointed out previously in the discussion of hydrostatics, a submarine operating on the surface behaves as does any surface craft. It is subjected to the same force phenomena when moving through the water. However, in considering the modern submarine, the operational requirements are such that the time spent on the surface is limited almost entirely to the time in port and in transit to deep water. These same operational requirements, as will be seen below, produce submarines whose total design of systems and configurations are wholly oriented to the three-dimensional, highly pressurized environment deep beneath the sea's surface.

Prior to World War II, the configuration of submarines was the result of a compromise between surface and subsurface operation. Because of their dependence on surface oxygen for cruising operation and their limited battery power for extensive submerged operation, their hull form was a concession to the existence of wave-making resistance. They were long and narrow, having an L/B ratio of about 11.5, with a center of volume approximately amidships. The bow configuration was a modified surface ship's bow, and there was considerable flat-deck surface with many unstreamlined appendages. The eddy-making resistance submerged was considerable, because the craft was essentially a mod-

ified surface ship capable of submerging for short periods.

The use of nuclear power has permitted the modern submarine to become a true subsurface ship. It is no longer dependent on the surface for oxygen to supply the engines. The crew breathes revitalized air, and the subsurface operation is limited more by personnel endurance. The present hull shapes are completely clear of appendages, except for the necessary control surfaces at the stern, the propeller, and the streamlined sail enclosure. The basic configuration of the bare hull is that of a body of revolution whose L/B ratios range from 8 to 11 for attack submarines and from 11 to 13 for ballistic-missile submarines.

Thus, the modern submarine's surface-resistance characteristics are completely subordinate to the speed and maneuverability requirements of submerged operations. The eddy resistance is reduced to a minimum and the total resistance submerged is almost completely frictional, except for eddy or form resistance, which is similar to that of aircraft.

The modern submarine experiences no wave-making resistance whatsoever when submerged more than 3 diameters from the free surface. This means that the total submerged resistance, including all of the frictional resistance of deck and sail as well as form and eddy drag, is only a comparatively small fraction of its total resistance at the same speed on the surface (figure 12-18).

In order to make powering estimates for submarines, the resistance components are usually divided into those for the bare hull (subscript BH) and those of the appendages (subscript AP). The following symbols are useful in calculating total resistance of a full-scale submarine by several different methods:

C_{FBH} = frictional-resistance coefficient of the bare hull from equation 11-21 or 11-23 (which gives essentially the same values for $R_n > 10^7$)

C_{RBH} = residual (eddy) resistance coefficient of bare hull, generally assumed independent of Reynolds number

C_{VBH} = viscous-resistance coefficient of bare hull, which is equal to the total-resistance coefficient for a deeply submerged bare hull

Thus,

$$C_{VBH} = C_{RBH} + C_{FBH} \quad (12-21)$$

or alternately,

$$C_{VBH}/C_{FBH} = 1 + 0.5 \frac{B}{L} + 3 \left(\frac{B}{L} \right)^3 \quad \text{(from equation 11-24)}$$

where B = maximum beam or diameter of the submarine
 C_A = model-ship correlation allowance or roughness allowance for full-scale resistance estimates made without model tests
 S_{BH} = wetted surface of bare hull
 $C_S = S_{BH}/\pi BL$ = ratio of bare-hull wetted surface to that of the outside of a cylinder with the same length and beam
 C_{VAP} = viscous-resistance coefficient of the various appendages, generally 1.5 to 2.5 times C_{VBH} , depending on the length, shape, and Reynolds number of the appendage
 S_{AP} = wetted surface of the various appendages, such as the fairwater (sail), rudder, and planes, etc., generally totaling 0.1 to 0.2 S_{BH} .

These definitions and symbols can be combined in

the equation for the effective horsepower of a fully submerged submarine.

$$\text{EHP} = \frac{1}{550} \rho v^3 \left[(C_{VBH} + C_A) S_{BH} + \sum C_{VAP} S_{AP} \right] \quad (12-22)$$

The required shaft horsepower can be found from equation 12-20.

$$\text{SHP} = \text{EHP} / \text{Propulsive Coefficient (P.C.)} \quad (12-23)$$

The most difficult problem in estimating the required shaft horsepower is in the measurement or estimation of the appendage resistance. Model tests of a complete submarine hull form have the problem of achieving a turbulent boundary layer on the rudder and planes that may be only 3 to 5 percent of the submarine length and have correspondingly lower Reynolds numbers. There is also some difficulty in scaling up measured propulsive

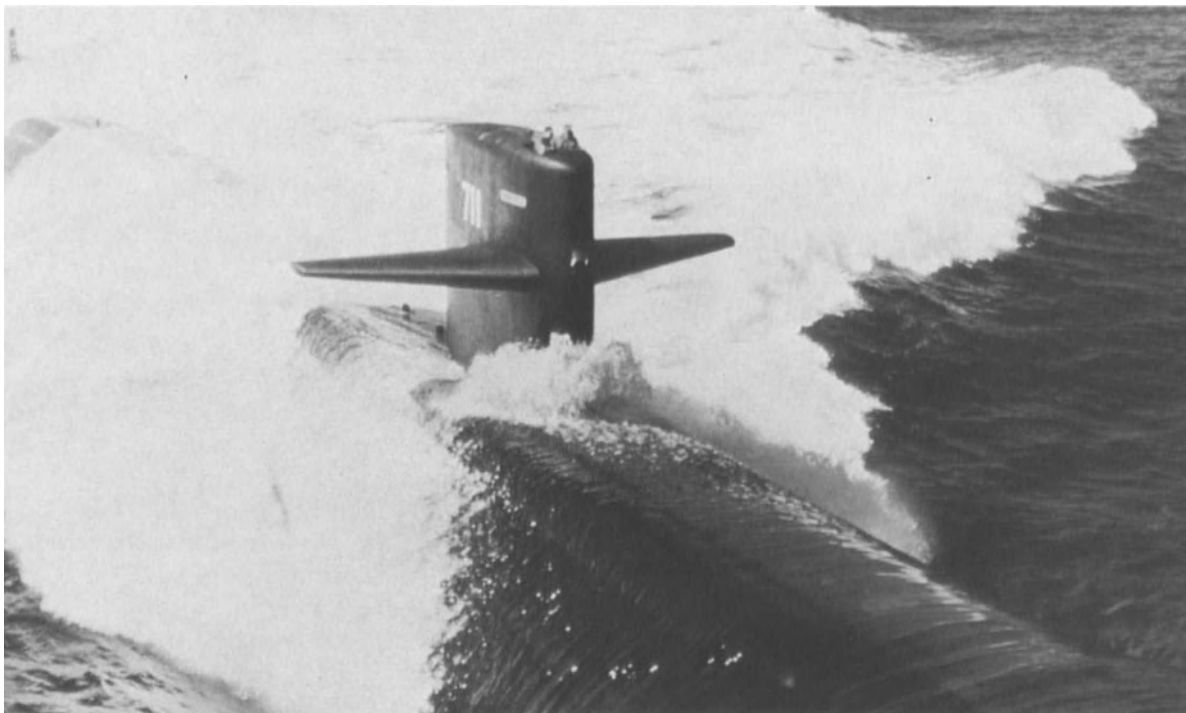
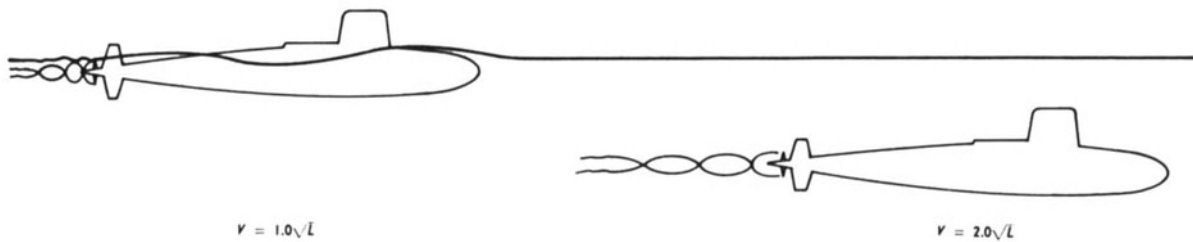


Figure 12-18. A navy attack submarine cruising on the surface creates a deep bow wave, which engulfs her forward quarters. This is a large expenditure of propulsion energy.

coefficients, which may vary from 0.7 to over 0.8 for a well-designed submarine propeller.

As an example of submarine powering calculations, typical data from *Jane's Fighting Ships* will be used, along with estimates of the appendage resistance for a nuclear attack submarine. An estimation procedure for the wetted-surface area for Series 58 (*Albacore*)-type hull forms is found from the approximation

$$C_S = 1.03(C_P)^{2/3} \quad (12-24)$$

Example 12-1: Estimate the EHP and SHP of a nuclear attack submarine traveling at 30 knots in 59°F seawater. The submarine is 260 feet long, has a maximum diameter of 32 feet, and a submerged displacement of 3,400 tons. Assume the total wetted surface of the appendages equals 20 percent of the wetted surface of the bare hull and that the average viscous-drag coefficient of the appendages is 1.8 times that of the bare hull. Use a roughness (correlation) allowance of .0002 and a *P.C.* of 0.75.

Solution: The prismatic coefficient of the bare hull is found from equation 3-2.

$$\begin{aligned} C_P &= \frac{\nabla}{LA_x} = \frac{35\Delta}{L\frac{\pi}{4}B^2} \\ &= \frac{4(35)(3,400)}{260\pi(32)^2} \\ &= 0.569 \end{aligned}$$

The wetted-surface coefficient is approximated from equation 12-24.

$$\begin{aligned} C_S &= 1.03(C_P)^{2/3} \\ &= 0.707 \end{aligned}$$

The wetted surface of the bare hull is found as follows:

$$\begin{aligned} S_{BH} &= C_S \pi BL \\ &= 18,490 \text{ ft}^2 \end{aligned}$$

The wetted surface of the appendages is estimated as

$$\begin{aligned} S_{AP} &= 0.2S_{BH} \\ &= 3,700 \text{ ft}^2 \end{aligned}$$

The Reynolds number of the bare hull is found as follows:

$$\begin{aligned} R_n &= \frac{vL}{\nu} = \frac{30(1.688)260}{1.279 \times 10^{-5}} \\ &= 1.03 \times 10^9 \end{aligned}$$

The bare-hull skin-friction coefficient is found from the 1957 ITTC line.

$$\begin{aligned} C_F &= \frac{.075}{(\log_{10} R_n - 2)^2} \\ &= .00153 \end{aligned}$$

The coefficient of viscous resistance for the smooth bare hull is found from

$$\begin{aligned} C_{VBH} &= C_{FBH} \left[1 + 0.5 \left(\frac{32}{260} \right) + 3 \left(\frac{32}{260} \right)^3 \right] \\ &= .00153(1.067) \\ &= .00163 \end{aligned}$$

The average viscous-drag coefficient of the appendages is estimated at

$$\begin{aligned} C_{VAP} &= 1.8C_{VBH} \\ &= .00294 \end{aligned}$$

The EHP estimate is found from equation 12-22.

$$\begin{aligned} \text{EHP} &= \frac{\rho v^3}{2(550)} [(C_{VBH} + C_A)S_{BH} + C_{VAP}S_{AP}] \\ &= \frac{1.9905(50.64)^3}{2(550)} [(.00163 + .0002)(18,490) \\ &\quad + .00294(3,700)] \\ &= 235 (33.84 + 10.88) \\ &= \boxed{10,500 \text{ horsepower}} \end{aligned}$$

The SHP estimate for a propulsive coefficient of 0.75 is

$$\begin{aligned} \text{SHP} &= \text{EHP}/P.C. \\ &= \boxed{14,000 \text{ horsepower}} \end{aligned}$$

The SHP estimate has many uncertainties associated with the values of C_{VBH} , C_A , and C_{VAP} . It is common to use a value of .0002 for C_{RBH} , which is twice that predicted by equation 11-24. Note that a change in C_A from .0002 to .0004 would increase the bare hull drag by 11 percent at 30 knots. The typical relative percentages of component resistance for submarine powering are given in figure 12-19.

Solving equation 12-22 for velocity as a function of the other variables, it is seen that speed varies as the cube root of the various factors for small changes in velocity. An uncertainty of 10 percent in any of the factors produces a 3.2 percent change in velocity (or one knot at 30 knots). Since the viscous-drag coefficients are functions of Reynolds numbers, it is useful to set up a computer program to calculate EHP and SHP as a function of velocity based on increments of,

say, one knot. Suggested homework problems include predicting maximum speed attainable from submarine data given in *Jane's Fighting Ships*.

Although submarines have higher ratios of wetted-surface area to displacement than cargo ships, this ratio is similar to that of high-speed displacement ships, such as destroyers and frigates, with similar L/B ratios. The advantage a submarine has over surface ships at high values of speed-length ratio is shown in figure 12-20, in which the values of EHP/ton are extrapolated well beyond the range of existing ships.

12-6 Power Sources: The Designer's Choice

To determine the amount of power for the required spectrum of speeds and operational missions is by no means the end of the power-selection problem. It has been described how the designer refines this determination through a model-test schedule. Before this time in the process of the design he must have decided what type of prime mover or engine combination would deliver this energy. When ships were first mechanically propelled, and perhaps for a century thereafter, there was little choice other than a reciprocating steam engine with a boiler. Near the turn of the century came the steam turbine and the diesel internal-combustion engine, followed closely by combined electric drive, both turbo electric and diesel-electric. Later in this century the gas turbine was advanced to the status of a practical power source. It is increasingly in use and has a bright future.

Perhaps the most sophisticated of marine power plants is the combination of a proper choice of two basic engines, such as gas turbine and diesel, either through two different propulsors or a single-propulsion system.

All of the power systems noted above have their own individual characteristics: their best operating-efficiency characteristics, their specific weights (pounds per horsepower), their characteristic fuel-consumption rates, and their power limitations. These factors are obviously most important to the designer, who must be intimately concerned with weight, volume, cruising range, and speed.

TYPES AND CHARACTERISTICS OF POWER SOURCES

All of the marine power plants currently in use and under contemplation for future use can be classified according to the following basic thermodynamic cycles:

1. *Rankine cycle*—Steam turbines of 2,000–100,000 SHP/unit
2. *Brayton cycle*—Gas turbines of 500–40,000 SHP/unit
3. *Otto cycle*—Reciprocating internal-combustion gasoline engines of 10–2,000 SHP/unit

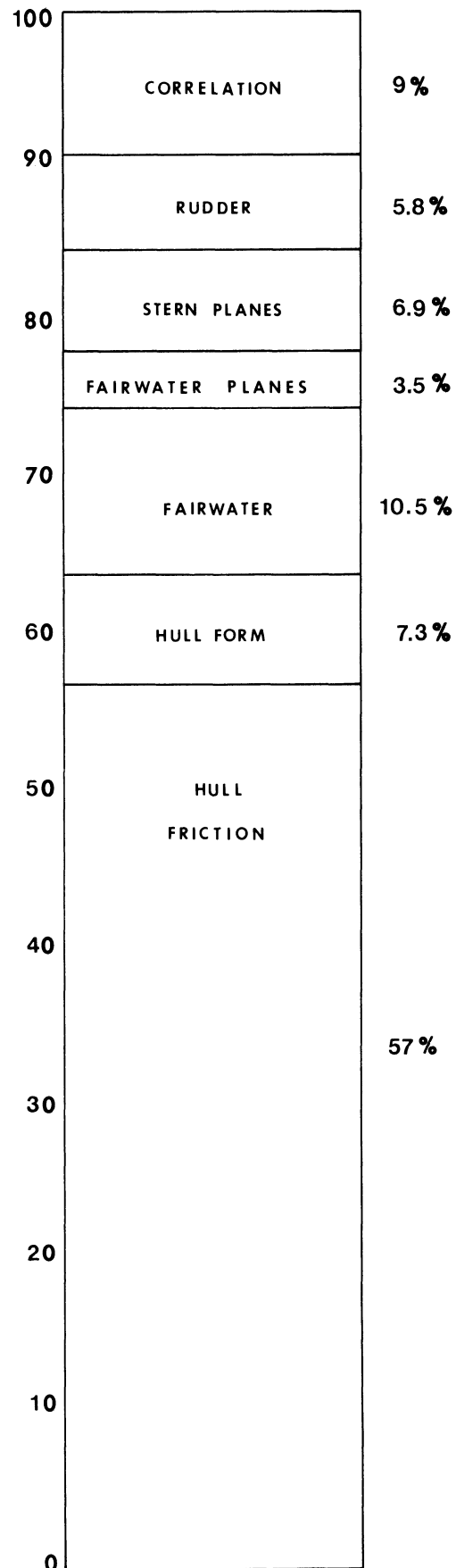


Figure 12-19. Typical component percentages of total submarine resistance

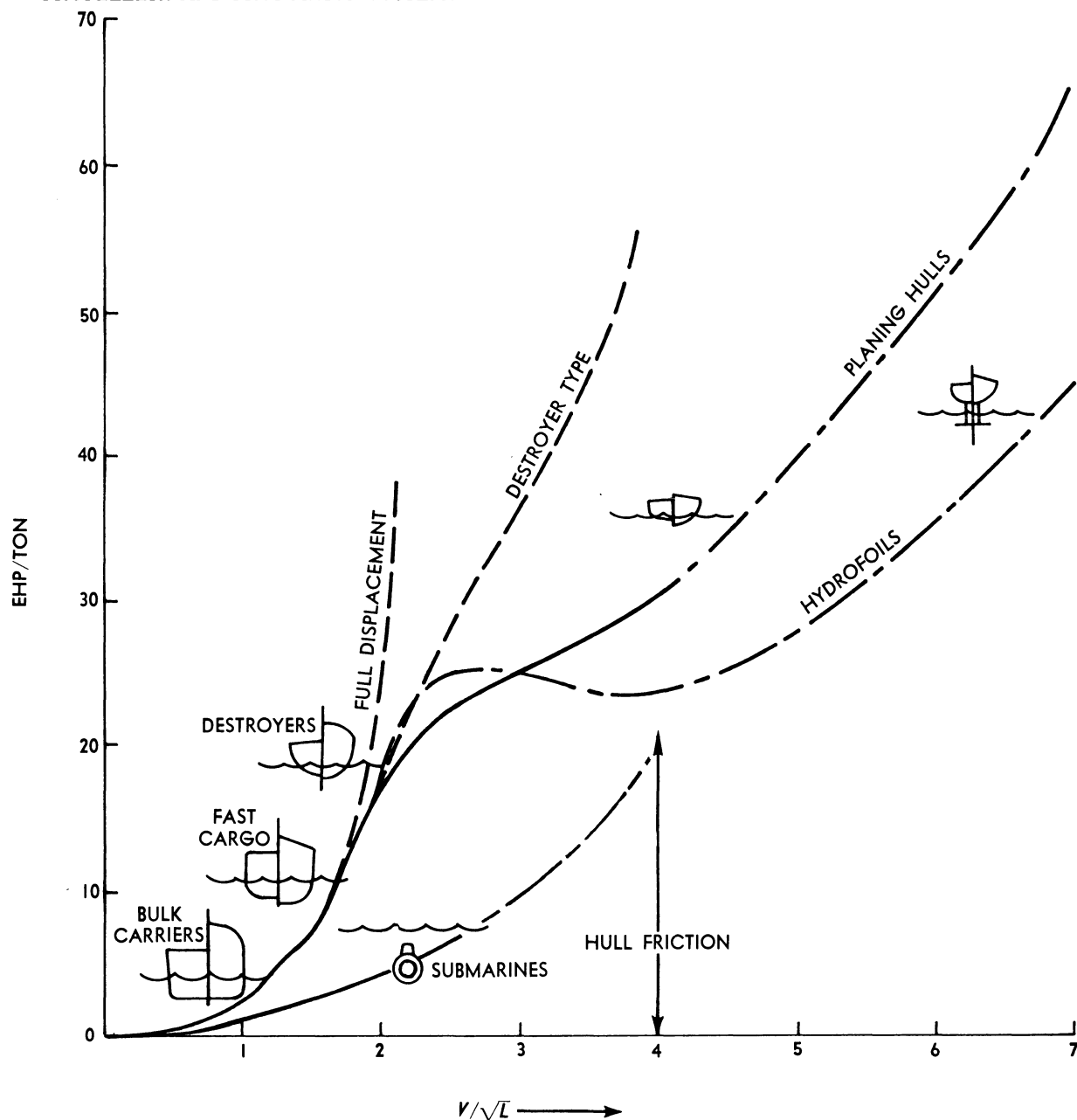


Figure 12-20. Speed-power trends for ship-type ranges

4. *Diesel cycle*—Reciprocating internal-combustion diesel engines with

- high speed (>1000 rpm) of 500–18,000 SHP/unit
- medium speed (200–1,000 rpm) of 500–27,000 SHP/unit
- low speed (direct coupled, <200 rpm) of 2,000–48,000 SHP/unit

For comparative purposes here, it is unnecessary to describe the above cycles further or to subdivide them into their various arrangements—closed, open, regenerative, etc. The inherent operating characteristics are common to all, and these characteristics contain the key to the choices available to the ship designer.

The above types and ranges are considered in the

context of surface craft. For submarines using nuclear fuel, the cycle is the steam (Rankine) cycle, generally of low to moderate pressure, using little or no superheat, and the engine powers are lower than those indicated. Because of the necessity for atmospheric air in all other cycles, the nuclear-fueled submarine is presently limited to the steam cycle, and its power must be designed to fit the submarine's requirements.

The performance parameters that interest the ship designer most in the early stages of design, when he is on the outer edges of his design spiral, are principally

- Horsepower ranges
- Specific fuel consumption (SFC), lb/hp hour
- Specific weight (sp.wt.), lb/hp
- Initial and life-cycle costs.

There are other performance parameters that must be investigated simultaneously with the above characteristics. Among these are the

1. Space and arrangement requirements
2. Maximum-to-continuous power ratio
3. Maintenance and repair requirements
4. Lubricating-oil consumption rates
5. Type of fuel required
6. Reversing capability
7. Operating personnel
8. Reliability
9. Ability to withstand damage (naval vessels).

It is apparent that when weighing the total effectiveness of these powering systems for a specific ship design, that problem immediately becomes one of optimization with all of the other ship-design factors. Such optimization in modern-design technique becomes a problem for computerized solution. This discussion must be more simply confined to the built-in performance ranges to be found and to be expected in the available marine-power-plant types.

In terms of total horsepower requirements, the selection is more straightforward, and for very large displacement-type ships requiring maximum horsepower, there is very little choice. However, as ship size may be limited, and speed and cruising range become more important with mission, the selection becomes more critical.

STEAM TURBINES

The steam turbine is well suited to the largest power plants, produces a uniform turning effort, burns the least expensive fuel, and is capable of relatively high efficiencies, long endurance, long life, long mean time between overhaul (MTBO), and reasonable maintenance. It has the inherent disadvantages of large space requirements, a large watch-standing force, slow response time, being slow in getting under way, intermediate energy conversion with its secondary working substance, mechanical nonreversibility, and high turbine speed. These latter two disadvantages necessitate a separate, low-efficiency astern turbine of reduced power and large, heavy reduction gearing to the propeller shafts. The speed reduction is needed to allow both the turbine and the propeller to operate in their most efficient speed range.

GAS TURBINES

Gas turbines are powering an increasing number of high-speed displacement ships and the majority of high-performance vessels. This is because they have the lowest ratio of machinery weight to horsepower (specific weight) of any of the marine power plants. Gas turbines are very reliable (the MTBO for an LM 2500 gas turbine used on many destroyers and frigates is over 12,000

hours), provide quick response after a cold start, and are adaptable to remote-control operation with a small watch-standing force. There is little maintenance between overhauls and most installations are designed to replace the entire engine, much like the aircraft jet engine from which it was derived. The disadvantages of gas-turbine propulsion include a large volume for the total plant, most of which is taken up by the inlet and exhaust ducting for the large quantities of air and exhaust gases. Although the gas turbine is reasonably efficient at the design power rating, it has a very high specific fuel consumption off the design rpm.

It also has no reversing capability, so controllable-pitch propellers are almost always found in gas-turbine-powered ships. To improve fuel consumption at cruising speeds in twin-screw vessels, one of the controllable-pitch propellers is often "feathered," or aligned with the flow, so the shaft does not rotate and all of the propulsion comes from the remaining shaft.

Gas turbines are perhaps most sensitive to atmospheric air temperature. Their power output will drop as inlet air temperature climbs. At the same time, the steam cycle's efficiency drops when the cooling water temperature in the condenser rises, which is also a function of its operating environment. It should also be pointed out that gas-turbine costs may be as much as 30 percent greater than those for steam power plants, since the gas turbine burns a higher grade fuel.

DIESEL ENGINES

The low-speed, direct-coupled diesel engine is the most efficient of all the marine power plants, with specific fuel consumptions as low as 0.29 lb/hp-hr. Since it is now available in horsepowers of up to 48,000, it is being used to power most new merchant ships requiring between 25,000 and 48,000 horsepower. Some steam-powered vessels are being converted to low-speed diesel operation because of the present high fuel costs. Both low-speed and medium-speed diesels can operate on heavy, residual fuel oils when properly treated and heated for injection. Medium-speed diesels approach the fuel efficiencies of low-speed diesels and are considerably lighter, smaller, and have lower initial costs. Both types are directly reversible, but controllable-pitch propellers are commonly used on medium-speed diesel installations, especially where maneuvering characteristics are important, as in Great Lakes ore carriers and ferryboats.

Both the diesel and gasoline (Otto cycle) engines are self-contained reciprocating engines with a multiplicity of moving and stationary parts. The engine frame referred to is the familiar engine block in the smaller engines as well as its equivalent in the large horsepower diesels that are universally built in cylinder modules.

The frame life is extremely long because of its nec-

essary rugged strength. It is difficult to measure the frame's rate of deterioration, because the only progressive deterioration normally evident is the rate of corrosion caused by seawater cooling, or where applicable, freshwater cooling. However, in the lifetimes of most marine diesels or gasoline engines, the replacement of worn parts and expired subsystems goes on at a steady monthly and annual rate. This even applies to the replacement of such major parts as cylinder liners, main bearings, valves, crankshafts, etc.

(INTERNAL COMBUSTION) GASOLINE ENGINES

Gasoline engines are primarily used in pleasure craft and high-performance craft, where their weight and initial cost advantage over diesels outweighs their higher fuel consumption, hazardous-fuel use, and high maintenance costs.

COMBINED SYSTEMS

It is evident at this point that some power plants have advantages not possessed by others and vice versa. No system can meet every requirement. Consequently, combined systems are necessary for the special craft that are required to keep at sea for long periods of cruising at comparatively low speed and low power, and yet that have the reserve muscle for high speeds of short duration.

CODAG

In recent years, the CODAG (combination diesel and gas turbine) system has been increasingly used in the most modern ships, from comparatively small patrol craft to larger destroyer-type escort craft, Coast Guard cutters, rescue craft, etc. The diesel engines offer good fuel economy for cruising and the gas turbines provide the lightweight power required for high speed.

COGOG

The COGOG (combined gas turbine or gas turbine) combines a high-power gas-turbine installation for burst speeds with a low-power gas turbine for cruising.

COGAS

In order to recover some of the waste heat that is exhausted by a gas turbine, the COGAS (combined gas turbine and steam) system has been proposed, in which steam is generated by a heat exchanger in the exhaust of the gas turbine and used to drive a steam turbine. This is a good solution for fuel economy in land-based installations of back-up power plants, but has yet to be proven as a marine power plant.

FACTORS AFFECTING ABILITY TO SATISFY FULL POWER REQUIREMENTS

There are a number of external factors that inhibit or reduce the capacity of the ship's propulsive system to satisfy the full power requirements. Among these, the most significant are

1. Shallow-water effects
2. Rough-water effects
3. Hull fouling
4. Injection temperature.

Both shallow-water and rough-water effects on resistance have been previously discussed, and their contribution to power prediction is accordingly applied at this stage.

Hull fouling by sea growth is dependent upon the type of bottom paint and the operational environment. These factors can be applied from available source data in terms of prepared tables and curves based on actual tests.

The effect of cooling-water injection temperature is simply a correction to SHP for variation to the steam plant's power cycle from that temperature used to compute the predicted plant power. As such, it must be taken into consideration, together with the above four factors, as variables to the ideal predicted conditions.

FINAL CONSIDERATIONS

The ultimate choice of the power system, as can be seen in the foregoing, depends upon a multiplicity of factors. For the larger vessels, in the absence of predetermined design direction of sister craft, the choice will increasingly depend upon the optimization procedures conducted through computer-aided analysis.

Problems

12-1. The following data was taken during a propeller test in the U.S. Naval Academy's circulating water channel:

Propeller diameter	8.8 inches
Propeller pitch	9.12 inches
Water velocity	9.6 feet per second
Shaft speed	20 revolutions per second
Thrust	34.4 pounds
Torque	52.5 inch-pounds

Determine

- a) slip ratio
- b) advance coefficient
- c) thrust horsepower
- d) thrust coefficient
- e) propeller horsepower
- f) torque coefficient
- g) open-water propeller efficiency

12-2. Write a computer program based on example

problem 12–1 that will calculate the EHP and SHP versus submarine speed in knots. Take the input data on the submarine length, diameter, and submerged displacement from *Jane's Fighting Ships*. Estimate the wetted surface of the appendages as a percentage of wetted

surface of the bare hull. Vary the roughness allowance and the propulsive coefficient to determine the sensitivity of these parameters. Set up a graphics output file or log the results in 1-knot increments from 5 to 40 knots.

Ship Motions in a Seaway

13-1 Introduction

Thus far the text has treated the operating environment as a flat and motionless sea. This was necessary in order to obtain analytical relationships where engineering calculations could be made to within one to 5 percent accuracy. However, as soon as a realistic environment is superposed on ship operations, the accuracy of the engineering calculations declines, since the analysis is based on statistical methods and probability theory. In studies of ship dynamics, it is possible to predict theoretically and to measure experimentally ship motions in response to two-dimensional simple harmonic waves to within 10 to 15 percent. However, ship motions in the real three-dimensional and frequently hostile sea can be predicted to within 50 percent with some degree of confidence. Studies on the relative merits of various ship-hull forms, concerning their *seakeeping* and *seakindliness* characteristics, cannot be taken as absolute predictions, but are only a measure of relative performance when operating in the assumed or simulated seaway (figure 13-1). In this context, *seakeeping* refers to a ship's ability to maintain normal functions at sea, and *seakindliness* refers to the quality of behaving comfortably and producing easy motions in a seaway. To fully appreciate the complexities of predicting ship motions in a seaway, the reader is referred to books such as *Dynamics of Marine Vehicles*, the previously mentioned *Proceedings of the International and American Towing Tank Conferences*, and various publications from the Society of Naval Architects and Marine Engineers.

13-2 Water in Motion

The wave motion of water in the open sea and in restricted waters is nondeterministic. To describe such motion analytically is extremely difficult and requires statistical methods. However, theories based on geometrical and mathematical analysis of periodic waves

have been advanced that closely approximate certain actual ocean-wave phenomena.

Ocean waves generate some of the most powerful forces of all existing natural phenomena. Because of their tremendous power and their complex action on ships and structures, these forces are also difficult to analyze in a quantitative sense. However, it will be useful here to describe in a more qualitative way the origin of ocean waves and the results of some observations about them.

PERIODIC WAVES

Certain types of gravity waves are periodic and are thus amenable to mathematical analysis. The transverse wave system generated by a surface ship is basically a periodic wave train with a well-defined phase speed (equation 11-12 for small-amplitude waves). Other periodic waves include so-called *regular waves*, generated in towing tanks and wave basins for purposes of ship-model testing and coastal-engineering studies. (Periodic irregular waves are generated in wave tanks by summing periodic component waves that are all multiples of some fundamental frequency.) Ocean swells generated hundreds of miles from where they are observed sometimes appear almost periodic, with small deviations about a *mean period*.

The simplest form of periodic wave is the common sinusoidal wave, found in *monochromatic* light waves, acoustic waves, and small-amplitude gravity waves. Such waves seldom occur naturally, but are studied extensively because of the reasonably good agreement between mathematical theory and experiment. For simple harmonic, long-crested (two-dimensional) surface gravity waves of small amplitude (linear or Airy waves), the following relationship holds (figure 13-2):

$$c^2 = \frac{gL_w}{2\pi} \tanh 2\pi \frac{h}{L_w} \quad (13-1)$$

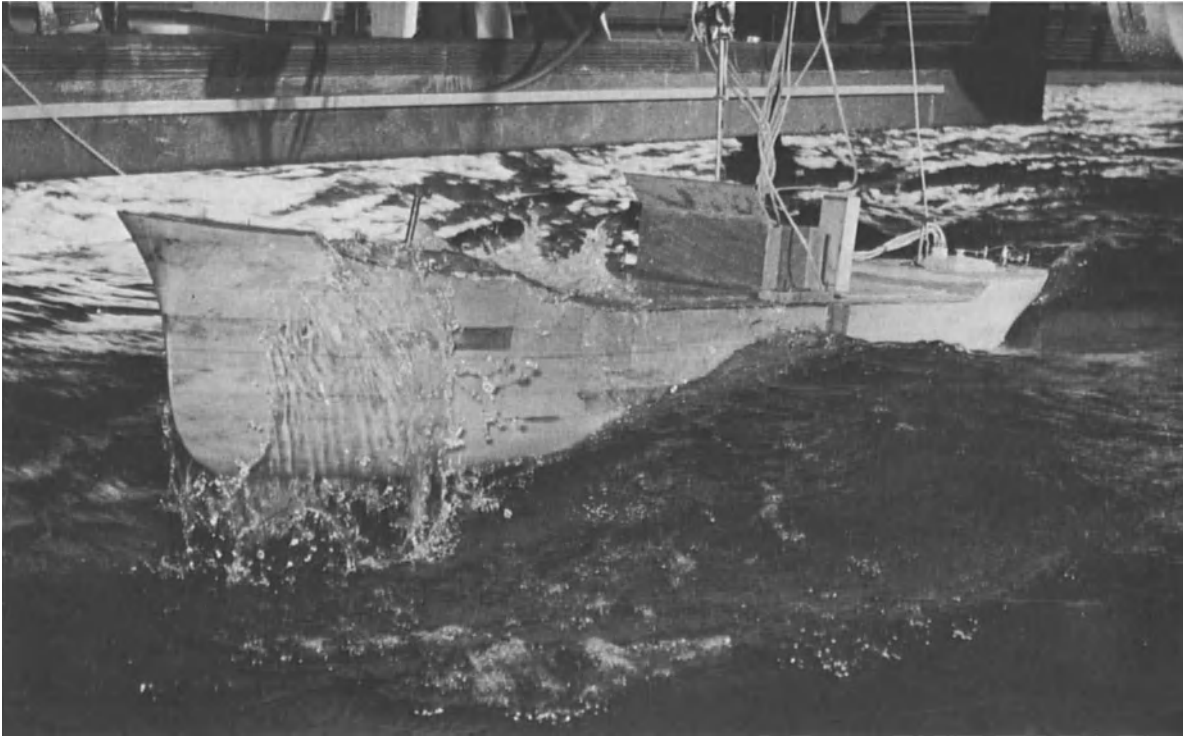


Figure 13-1. A ship model in irregular waves. (Courtesy of Netherland Ship Model Basin.)

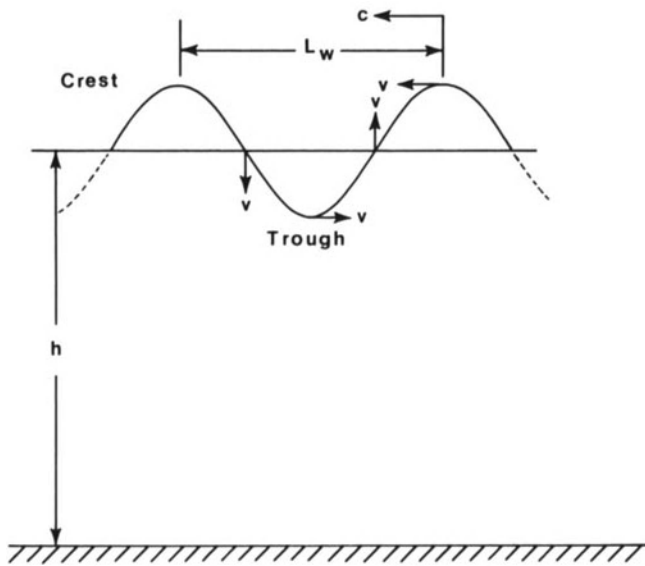


Figure 13-2. Sinusoidal regular waves

The period T of a sinusoidal wave is found from the time it takes the crest of a wave to travel one wavelength; that is,

$$T = \frac{L_w}{c} \quad (13-2)$$

The period of a given monochromatic (sinusoidal) wave is constant, regardless of the changes in wavelength and phase speed with water depth, as indicated in equation 13-1.

Deep-water waves are defined as those for which $\tanh 2\pi h/L_w$ is approximately equal to one, which occurs for water depths greater than half of the wavelength ($h/L_w > 1/2$). For these conditions,

$$c = \frac{L_w}{T} = \sqrt{\frac{gL_w}{2\pi}} = 2.26\sqrt{L_w} \quad (13-3)$$

$$= \frac{gT}{2\pi} = 5.12T$$

$$L_w = cT = \frac{2\pi}{g} c^2 = 0.195c^2 \quad (13-4)$$

$$= \frac{g}{2\pi} T^2 = 5.12T^2$$

where L_w = wave length (ft or meters)
 h = water depth (ft or meters)
 g = gravitational acceleration
 c = phase speed (velocity) of the wave front in ft/sec or meters/sec (not the particle velocity of the water)
 $\tanh(x)$ = hyperbolic tangent of x
 $= \frac{e^x - e^{-x}}{e^x + e^{-x}}$

$$T = \frac{L_w}{c} = 0.195c = 0.441\sqrt{L_w} \quad (13-5)$$

The phase speed of a simple harmonic wave front is greater than the actual water-particle velocities that travel in circular orbits of exponentially decreasing radius with depth, as illustrated in figure 13-3. (In intermediate and shallow water, where the influence of the bottom is felt on the particle motion, the orbits become ellipses that flatten out and extend to the bottom.) The particle velocity at the surface for a small-amplitude wave is found from

$$v = \frac{\text{distance traveled}}{\text{wave period}} = \frac{\pi H}{T}$$

where H = wave height

$\cdot$ = double amplitude of a sinusoidal wave.

Thus,

$$v = \frac{\pi H}{L_w/c} = \sqrt{\frac{g\pi}{2}} \frac{H}{\sqrt{L_w}} = 7.15 \frac{H}{\sqrt{L_w}} \quad (13-6)$$

and

$$\frac{v}{c} = \frac{\pi H}{L_w} \quad (13-7)$$

= maximum slope of sinusoidal wave

The validity of these equations breaks down, depending on the accuracy desired between *small amplitude* ($H/L_w \ll 1$) and breaking waves, for which $v \geq c$. Since waves generally break for values of H/L_w between 1/10 and 1/7, nonlinear effects must be considered in dealing with waves of *finite amplitude*. Several methods are available, including *trochoidal wave theory*, long used by naval architects (Comstock 1967, chapter 9), and higher order Stokes theories, used by physical oceanographers (Kinsman 1965) and ocean engineers (McCormick 1973). The trochoidal wave pictured in figure 13-3 is characterized by relatively sharp crests and longer, flatter troughs than those of a sinusoidal wave. It more closely resembles observed large-amplitude periodic waves, such as ocean swells and large regular waves in a towing basin.

In this diagram, a particle of water on the surface rotates in the largest circles. Such a particle will be at the top of its circular orbit when the crest of the wave is, at that instant, passing it. Likewise, the same particle will be at the bottom of its orbit when the trough of the wave is passing. It is in fact this uniform rotational velocity of each adjacent particle on the surface that produces the wave motion. At each successively deeper layer, the orbits of the particles become successively smaller in radii, thus producing waves in these layers of decreasing amplitude. The orbital radii approach an

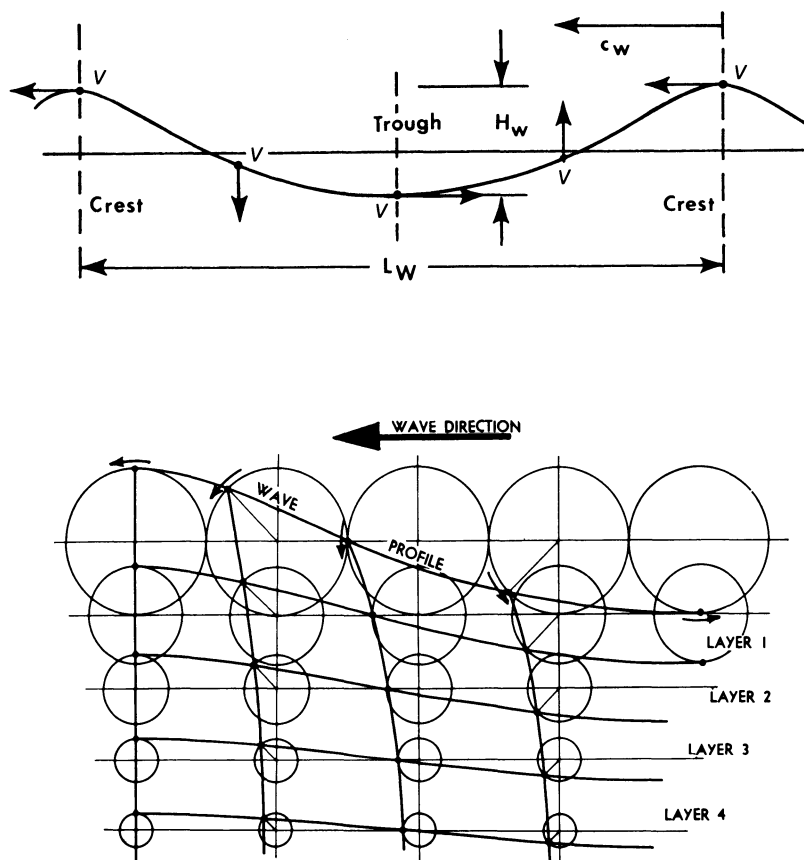


Figure 13-3. Structure of a trochoidal wave

infinitesimal value as the depth is further increased, and the resulting wave disturbance at some finite depth below the surface becomes negligible.

Geometrically, a point on a circle rolling on a horizontal line describes a curve called a *cycloid*. A point on a concentric circle of smaller radius within this first circle describes a *trochoid*. In figure 13-3, the points on adjacent orbits that show the relative position of the particles in the orbits are connected by a curve. This curve is a trochoid. The trochoidal wave generated may be steep or shallow, depending upon the ratio of the diameter of the generating circle to the wavelength. This trochoidal-wave form corresponds very closely to actual observed periodic waves, and its exact geometrical structure allows analysis of the motion and the forces applied to a ship floating in waves.

A typical example of its application has already been described in chapter 5, where hogging and sagging stresses of a ship were analyzed on a trochoidal wave of height $1.1\sqrt{L_w}$ (see figures 5-18 and 5-19).

WAVE ENERGY

It was shown in the foregoing section that in deep water each wave particle moved in a circular orbit at a constant angular velocity. Consequently, each particle of water will have a constant kinetic energy, or

$$\frac{m}{2} (\omega r)^2$$

where m = the mass of the particle
 ωr = the instantaneous velocity.

For a whole wave stratum of a single wavelength, including one trough and one crest, the energy is half potential and half kinetic. By integration of the above, the total energy may be reduced to an expression that, when simplified, gives

$$E = \frac{\rho g H^2}{8} \quad (13-8)$$

per unit of surface area, or

$$E = \frac{\rho g L_w H^2}{8} \quad (13-9)$$

per unit of wave breadth for one wavelength,

where ρg = specific weight of the water (lbs/ft³)
 L_w = wavelength (ft)
 H = wave height (ft) = vertical distance from trough to crest
 E = energy per foot of wave breadth.

According to the above expression, a 600-ft-long wave whose height is $L/20$ would have an energy of about 2,000 ft-tons per foot of width.

Most ships that attempt to operate in seas whose energy is of this magnitude obviously cannot control it (see figures 13-4 and 13-5). Actual instrument tests have measured wave forces as large as 6,000 pounds per square foot.

By referring again to the equation above, it will be seen that the energy necessary to produce a wave varies approximately in proportion to the square of its height; thus, a wave twice as high as another of the same length and speed requires four times the energy to sustain it. This is an important and interesting facet of wave theory when applied to the wave-making resistance of a ship. It has undoubtedly been noticed by anyone who has observed ships moving close at hand that some vessels make a deeper and more pronounced wave system than others. As shown in the above equation, the amount of energy expended in producing a wave of a given length and speed varies in proportion to the square of its height. This visible, deep wave along the side of a tugboat or cargo vessel is evidence of a comparatively large amount of power expended at a fairly low speed. A destroyer or cruiser moving at the same low speed would have an almost imperceptible transverse wave of the same wavelength. Thus, ships with large displacement-length ratios or large beam-length ratios produce large wave systems that represent the energy required for moving large masses of water per unit of time.

13-3 Wind-Generated Waves

Sea waves are primarily the result of the transference of the kinetic energy of local winds to the sea surface. The local winds are a result of complex interactions between thermal-energy transfer from the sun and the prevailing global weather patterns of the earth. These interactions can produce everything from calm days with only occasional puffs of wind to full-blown ocean storm systems that threaten everything in their paths. A mild breeze passing over a broad, flat expanse of water will first generate small ripples (*capillary waves*) on the surface as a result of the combined effects of fluid friction and the irregular nature of pressure variations in a turbulent gust of wind. If the gust is short lived (*cat's paws*), the capillary waves will die out from internal friction. If the breeze is steady and of sufficient duration, the waves will begin to grow in height and length and eventually become gravity waves that can travel far beyond the reach of the wind.

The height and length of the waves depends on the wind velocity, the intensity of any gusts associated with the mean wind speed, the duration of the wind, and the *fetch*, which is the horizontal extent of the water surface over which the wind blows. *Fetch-limited waves*



Figure 13-4. Long, cresting seas of the US West Coast. The height of the breaking sea on this ship's bridge can be easily gauged, since the height of the ship's bridge deck is 33 feet above her waterline. (Courtesy of the San Francisco *Examiner*.)

are generated in restricted waters, such as bays, lakes, and harbors; they are relatively short, choppy waves, limited in length by the fetch and limited in height by the maximum wave steepness sustainable before wave breaking occurs. The appearance of *white-caps* indicates that the rate of wave-energy buildup is being limited by turbulent energy transfer in breaking waves and the resulting internal friction that dissipates the kinetic energy to heat.

If the weather system is large enough on the open ocean, very long waves can be generated. Therefore the wave-breaking steepness limitation does not prevent the occurrence of very high waves. Eventually a balance between wave-energy production and dissipation can result in a *fully aroused* or *fully developed* sea, possessing certain steady-state statistical characteristics that may be described analytically.

Nearly all wind-generated waves are *short-crested waves*, in which the component waves advance in various directions as indicated by the relatively narrow width of most whitecaps. The periodic waves discussed in the previous section are generally assumed to advance only in one direction and are therefore two dimensional. For both types of waves, however, the longer wave components travel faster than the shorter components, as approximated by equation 13-3. So-called *wave-wave interactions* can result in very steep *rogue waves* in the open ocean, which are the result of larger

waves overtaking shorter waves and possibly combining wave heights of many of the waves in a particular group. Wave-group interactions of this type are not well predicted by probability theory, and the generation of extreme waves is still little understood.

Once a storm has subsided, the sea waves it generated fan out in many directions, with the longer waves apparently traveling farthest and gradually reforming into fairly long-crested, semiperiodic swells, encountered hundreds of miles from low-pressure storm centers. These swells slowly diminish in height as they radiate from their source and eventually are barely perceptible on the open ocean. As they travel into shoaling waters, however, they again increase in height and appear as the familiar breaking waves on surfing beaches around the world. Were it not for the shoaling process and the resulting wave breaking, the wave energy of the world's oceans would be many times what it is.

13-4 Descriptions of Irregular Waves

Irregular waves may be described in the *time domain* by a record of the instantaneous wave elevation measured by a wave height gauge of some type, as illustrated in figure 13-6. They may also be described in the frequency domain by using what is called the *Fourier transform* to map the time-domain record into a *frequency*



Figure 13-5. A tanker's bow meets the oncoming sea. The sea's force is evident as the vessel fails to rise out of the trough adequately. The ship is the 50,000-ton tanker *Marita* of Ocean Oil Carriers, Inc. (Courtesy of Bernard Benson.)

spectrum. The relationship between the time-domain representation and the frequency-domain representation is shown in figure 13-7. Artificial irregular waves composed of discrete harmonic components are represented in the frequency domain by the component spectral densities, which are proportional to the coefficients in a Fourier series expansion of the periodic time-domain function. However, ocean-wave-elevation data never repeat and cannot be represented by a Fourier series. The nonrepeating wave train can be represented by a statistical estimate of the continuous spectrum, as shown in figure 13-8. Since it is possible for several different wave records to have the same spectrum, and it is not possible to detect any information concerning the directional properties of the waves from a single-point measurement, one is faced with an incomplete description of the wave system. Nevertheless, it is useful to specify certain parameters such as the *significant wave height* $H_{1/3}$, which is the average apparent height of the third highest waves in an irregular wave pattern. The significant wave height is thought to be what most observers would estimate as an average wave height from visual observations. It is defined more precisely in statistical wave-height measurements and

is somewhat correlated with sustained wind speed for fully developed seas. Table 13-1 lists the commonly accepted correlation between sea state, significant wave

Table 13-1. Sea-State Chart for Worldwide, All-Season Naval Operations

<i>Sea state</i>	<i>Significant wave height (feet)</i>	<i>Sustained wind speed at 32.8 feet (knots)</i>	<i>Percent frequency of occurrence</i>
1	0 to 1.9	0 to 10	7.5
2	1.9 to 4.1	10 to 14	25.0
3	4.1 to 5.7	14 to 17	12.5
4	5.7 to 7.4	17 to 20	8.0
5	7.4 to 13.0	20 to 25	39.5
6	13.0 to 20.8	25 to 32	6.6
7	20.8 to 40.3	32 to 44	<0.9
8	40.3 to 61.6	44 to 55	~0.0

Note: The percent-frequency-of-occurrence statistics contained here were based on the best set of historical data currently published. Even so, major portions of the North Pacific are not covered in the data set.

The most likely occurrence is sea state 5, with sea state 2 being the second most likely. The apparent discontinuity in these two occurrences is due, at least in part, to the uneven spacing of the wave-height ranges.

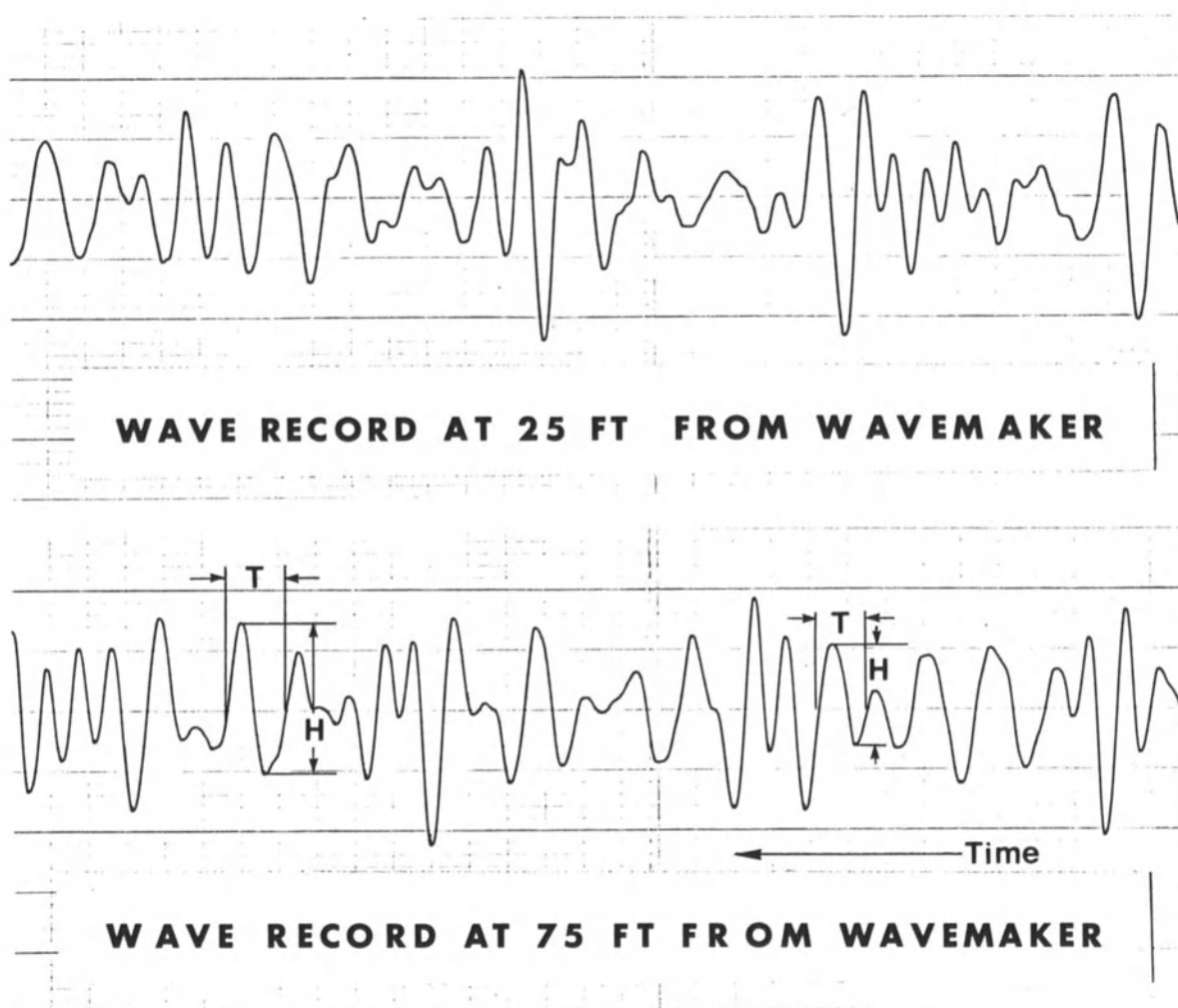


Figure 13-6. Typical record of irregular sea and definitions of apparent wave heights and periods

height, sustained wind speed, and percent frequency of occurrence on a worldwide basis for all-season operation.

13-5 Ship Motions

Ship motions in a confused, three-dimensional sea are very complex but can be broken down into six degrees of freedom relative to three mutually perpendicular coordinate axes through the ship's center of gravity, as shown in figure 13-9.

The three translational motions are

1. *Surge*—The longitudinal disturbance fore and aft along the ship's track superimposed on the ship's forward velocity

2. *Sway*—The lateral disturbance along the y-axis as the port and starboard drift

3. *Heave*—The vertical disturbance caused by the imbalance between the weight of the ship and the instantaneous changes in the buoyant force resulting from wave action.

The three rotational motions are

4. *Roll*—The transverse oscillatory rotation about the ship's longitudinal axis (as distinct from any non-oscillatory heel or list)

5. *Pitch*—The longitudinal oscillatory rotation about the ship's transverse axis (as distinct from any non-oscillatory trim)

6. *Yaw*—The rotation about the ship's vertical axis.

All or any of these motions may coexist in a given short time period, one being superimposed on another, resulting in a complex motion that is difficult to describe. For this reason, studies are often made at a particular heading in which some of the degrees of freedom are suppressed. In head seas, for example, pitch, heave, and surge are apparent, but there is little sway, roll, and yaw. Beam seas produce primarily roll, heave, and sway. The importance of minimizing any of the above motions depends on the mission of the ship and the sea conditions in which she is expected to maintain operational capabilities.

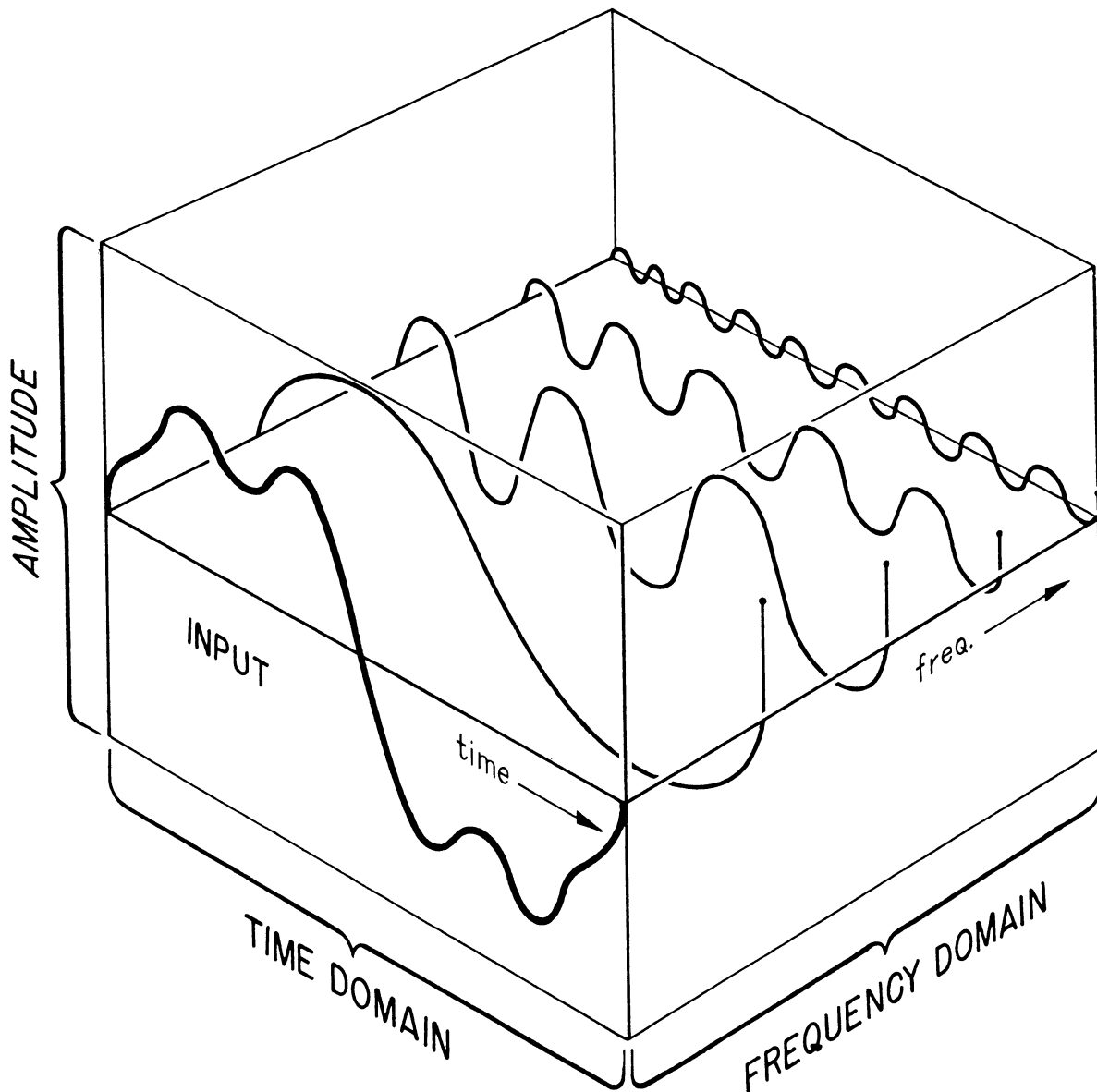


Figure 13-7. Time domain and frequency domain representations of a periodic irregular wave. (Courtesy of Hewlett Packard.)

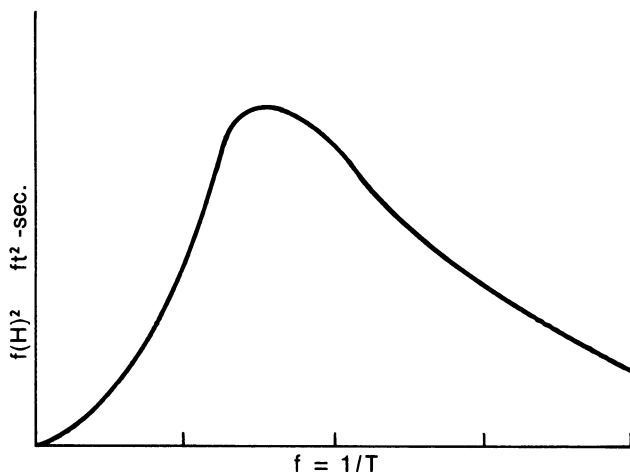


Figure 13-8. Continuous sea spectrum graphically relates functions of wave height and period

HISTORICAL ASPECTS

Since sailing vessels are unable to sail directly into the wind, they seldom encounter straight head seas. Square riggers most often choose routes that let them sail in beam winds to following winds. In beam winds, the sails act as large damping devices, so roll motions are restricted. Powered vessels lack the steadying influence of sails, so they tend to roll more readily and to larger angles. In following winds and seas, both types of vessels suffer from loss of rudder control during surges down the front of a wave (discussed in the next chapter). Only powered vessels venture into head seas, and this is where the added resistance of the wave forces, as well as the potential for slamming and shipping water, is maximized. When the first steamships with auxiliary sails attempted to follow more direct routes in world

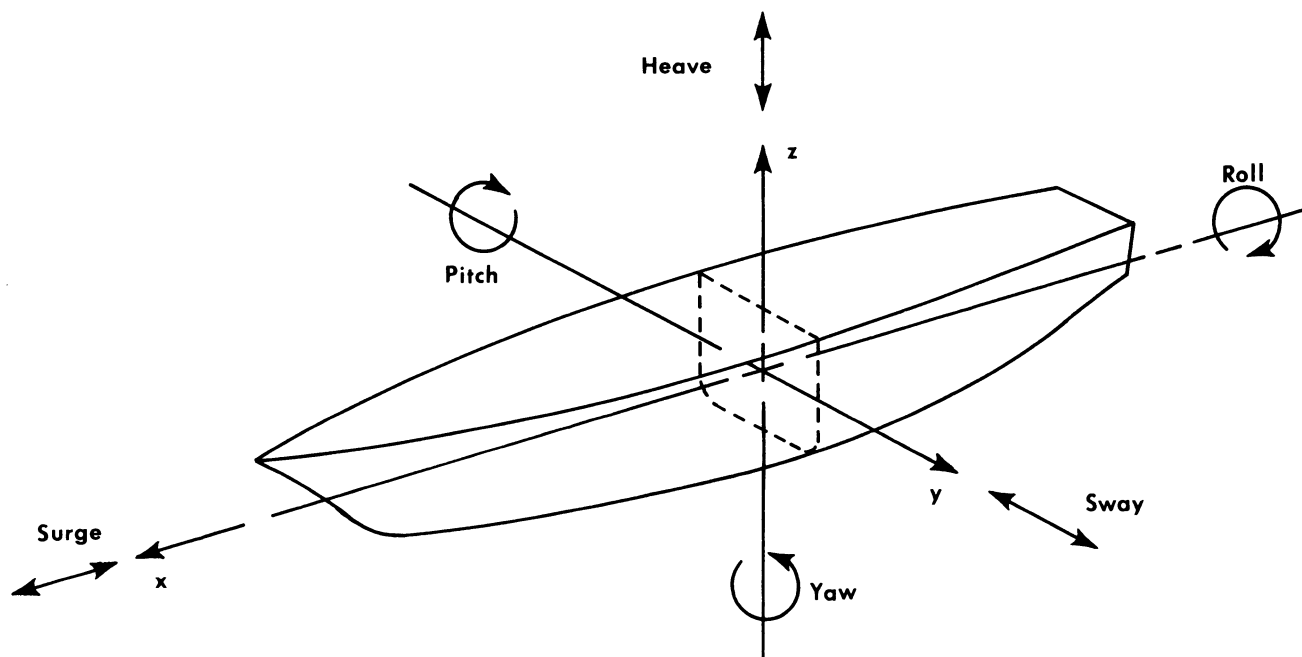


Figure 13-9. Ship motions

trade, they were battered by head seas and exhausted their coal bunkers. Following the sailing routes, which were considerably longer, they were able to utilize the combination of sail and steam power to break the sailing records by powering through the “doldrums.” The more powerful steamers of the late nineteenth century could travel more direct routes, and the high-powered vessels of today must occasionally reduce speed to avoid structural damage from bottom slamming and shipping water over the bow. For these reasons, ship routing is still used to avoid storms whenever possible.

13-6 Analysis of Uncoupled Ship Motions

The analysis of ship motions is complicated by the fact that several of the motions are always coupled, especially for ships that are asymmetrical, fore and aft. For these hull forms, the center of flotation is not lined up with the longitudinal center of buoyancy, so the restoring force forms a couple with the gravitational force. Figure 13-10 illustrates the coupling between pitch and heave for a transom stern vessel. In the top trace, the center of gravity of the small boat was depressed into the water and released. The resulting heave recorded shows the damping effects on the oscillation, and the pitch record shows the coupling effects between pitch and heave. In the bottom trace, the bow was depressed and released and the pitch motions also excited the heave motions.

Rather than treating the motions as coupled, a simplified analysis of the uncoupled motions will still yield

useful insight into the nature of the ship response. The format for this analysis is that used to describe the motions of a mass suspended from a spring, with and without viscous damping, as illustrated in figure 13-11. The basic equation for the motion is the familiar

$$\Sigma F = ma$$

or, exciting force minus damping force minus restoring force equals mass times acceleration. Using z as the displacement in the vertical direction,

$$\begin{aligned} \dot{z} &= dz/dt = \text{vertical velocity} \\ \ddot{z} &= d^2z/dt^2 = \text{vertical acceleration} \\ f(t) &= \text{exciting force} \\ c &= \text{damping coefficient} \\ k &= \text{spring constant} \\ m &= \text{mass of the suspended weight.} \end{aligned}$$

Thus,

$$f(t) - c\dot{z} - kz = m\ddot{z}$$

or, rearranging,

$$m\ddot{z} + c\dot{z} + kz = f(t) \quad (13-10)$$

Three cases are generally considered.

1. Undamped free vibration, with $c = f(t) = 0$
2. Damped free vibration, with $f(t) = 0$
3. Forced vibration with harmonic excitation, with $f(t) = A \sin \omega t$

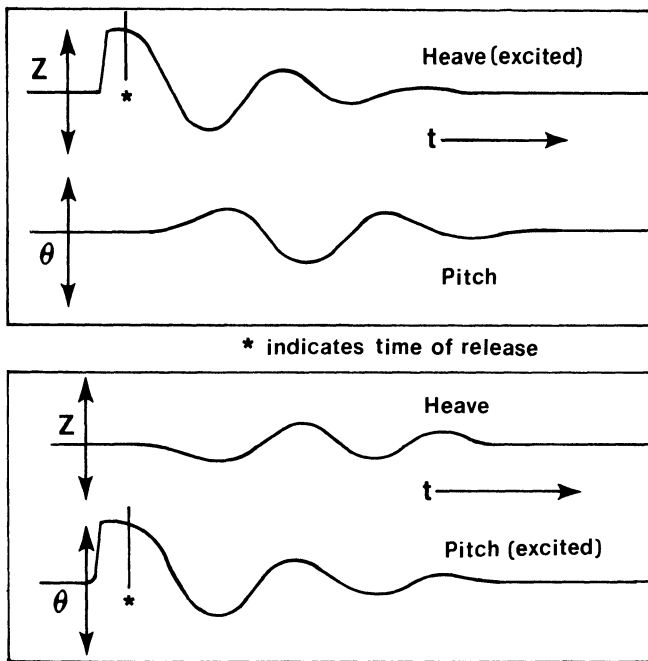


Figure 13-10. Coupled pitch and heave motions

For all three cases, a reference frequency of *undamped natural frequency* is defined, which is part of the solution to the undamped free-vibration case.

$$m\ddot{z} + kz = 0 \quad (13-11)$$

To convert this expression to the “wave equation,” let

$$\omega_n^2 = \frac{k}{m} \quad (13-12)$$

Then

$$\ddot{z} + \omega_n^2 z = 0 \quad (13-13)$$

which has a general solution of

$$z(t) = A \cos \omega_n t + B \sin \omega_n t$$

where ω_n = undamped natural frequency.

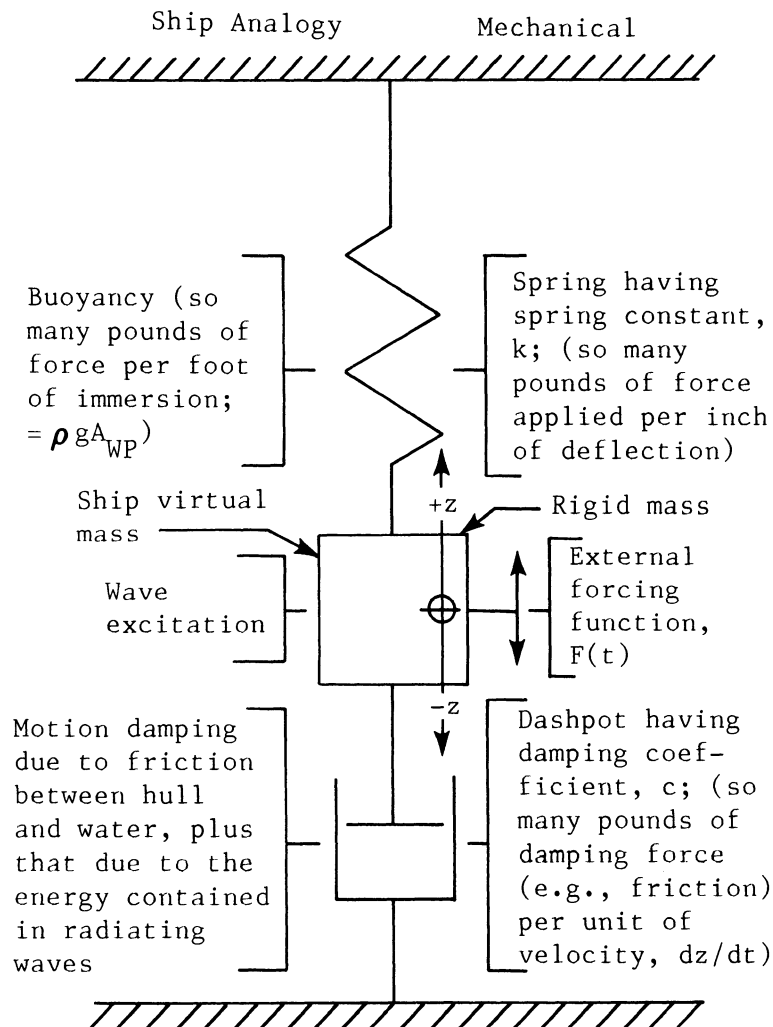


Figure 13-11. Oscillation of a spring-mass system

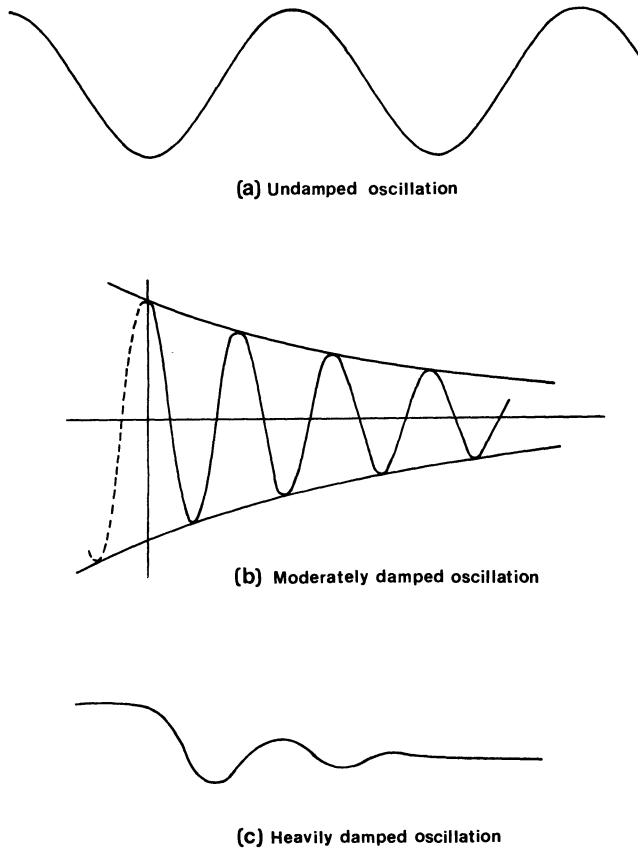


Figure 13-12. Damped and undamped oscillations: (a) undamped oscillation (b) moderately damped oscillation (c) heavily damped oscillation

The equivalent *undamped natural period* is

$$T_n = \frac{2\pi}{\omega_n} = 2\pi \sqrt{\frac{m}{k}} \quad (13-14)$$

If the mass in figure 13-11 is released from some elevated or depressed position relative to its rest position, and if the damping dashpot is absent, it will oscillate indefinitely at the undamped natural frequency ω_n , as illustrated in figure 13-12(a).

If some damping is present, the mass will oscillate with decreasing amplitude and a slowly changing period, as illustrated in figure 13-12(b). If heavy damping is present, the mass may oscillate only once or twice, as in figure 13-12(c). If *critical damping* is present, the mass will return to its original position without oscillation.

If periodic excitation $f(t)$ is now added, the motions of the mass may be either amplified or reduced relative to the exciting function, depending on the ratio of the excitation frequency to the undamped natural frequency (called the tuning factor Δ) and the ratio of the damping coefficient to the critical-damping coefficient,

as shown in figure 13-13. Note that heavy damping reduces the apparent resonant frequency below that of the undamped natural frequency.

HEAVE MOTIONS

To relate this mechanical analogy to the heave motions of a ship, note that the spring constant for a ship is proportional to the area of the waterplane or the tons-per-inch immersion (TPI),

$$k = \text{TPI} \times 12 \text{ in/ft}$$

and the mass undergoing oscillation equals the mass of the ship plus the *added mass* of the water accelerated by the motion of the ship. The added mass of water is usually expressed as a percentage of the mass of the ship x_A .

$$m = \frac{\Delta}{g} (1 + x_A)$$

Thus,

$$T_{nz} = 2\pi \sqrt{\frac{\Delta(1 + x_A)}{12g\text{TPI}}} \quad (13-15)$$

Equation 13-15 is difficult to verify experimentally because heave motions are heavily damped, and thus the concept of a natural period in heave is somewhat questionable. If the motions were lightly damped, the added mass coefficient x_A could be determined by measuring

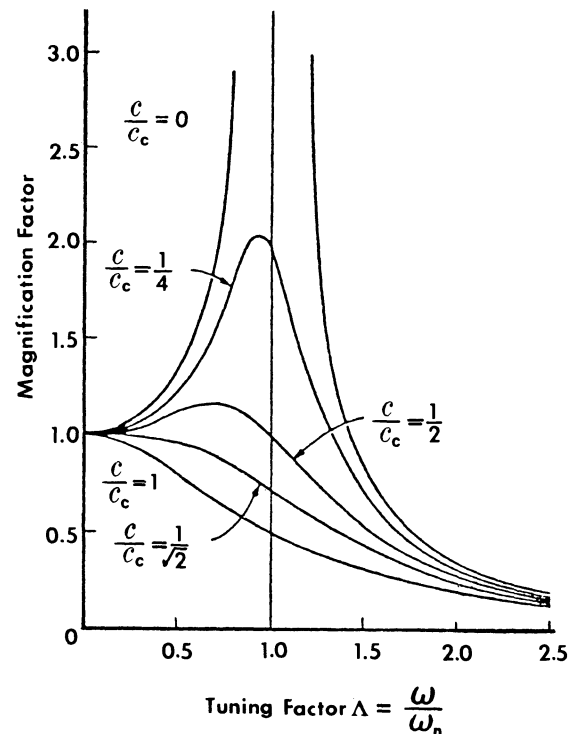


Figure 13-13. Magnification factor for uncoupled heaving or pitching in waves

the natural response period of the heave motions. Such is not the case, and the coupled pitch motions further complicate the estimation of added mass. Theoretical methods may give some guidance in establishing the value of the added mass for certain hull geometries.

The primary significance of heaving motion alone is the effect of acceleration on the apparent mass of the ship. Recalling that transverse stability is expressed as the product of the ship's displacement and the righting arm, it is apparent that in a seaway, where the ship is subjected to heaving motion, the stability may be both increased and decreased periodically as the acceleration becomes maximum in the positive and negative sense. For example, a ship lifted upward by a wave would, at the topmost position in this movement, have the minimum apparent gravity and, consequently, have at this instant the least resistance to heeling moments (depending on the instantaneous waterplane area).

UNDAMPED ROLL AND PITCH MOTIONS

For undamped oscillations in roll and pitch, the equations of motion for angular momentum equivalent to equation 13-11 for a ship are

$$I(1 + x_A)\ddot{\phi} + \Delta\overline{GM}\phi = 0 \quad (13-16)$$

where I = moment of inertia about the appropriate axis
 x_A = added mass coefficient for the particular motion
 $\ddot{\phi}$ = angular acceleration
 $\Delta\overline{GM}\phi$ = righting moment (limited to $\phi < 10^\circ$ converted to radians, so that $\overline{GZ} = \overline{GM}\phi$).

It is useful to relate the moment of inertia to the radius of gyration k_x or k_y from equation 2-38.

$$I_x = mk_x^2 \text{ for roll motions}$$

$$I_y = mk_y^2 \text{ for pitch moments.}$$

Thus,

$$\ddot{\phi} + \frac{\Delta\overline{GM}}{\frac{\Delta}{g}(1 + x_A)k_x^2} \phi = 0 \quad (13-17)$$

Define the *undamped natural frequency* in roll as

$$\omega_{n\phi} = \frac{\sqrt{\Delta\overline{GM}}}{(1 + x_A)k_x} \quad (13-18)$$

so equation 13-17 becomes the equation for simple harmonic motion,

$$\ddot{\phi} + \omega_{n\phi}^2 \phi = 0$$

with an undamped natural period in roll of

$$T_{n\phi} = 2\pi k_x \sqrt{\frac{1 + x_A}{g\overline{GM}}} = 1.108 k_x \sqrt{\frac{1 + x_A}{\overline{GM}}} \quad (13-19)$$

Since roll motions are lightly damped, this expression can be used experimentally to estimate the roll gyradius, which would include the effects of added mass. If the roll gyradius is known from the weight distribution, the expression could be used to estimate the added mass coefficient in roll.

For pitching motions, ϕ is replaced by θ , k_x by k_y , and $\overline{GM}$ by $\overline{GM}_L$. Equation 13-19 becomes

$$T_{n\theta} = 1.108 k_y \sqrt{\frac{1 + x_A}{\overline{GM}_L}} \quad (13-20)$$

Pitching motions are also heavily damped, so the experimental verification of equation 13-20 is difficult.

13-7 Roll Motions

In a previous discussion of stability characteristics, a ship of a comparatively large $\overline{GM}$ was found to be more uncomfortable in a seaway than a ship of a smaller $\overline{GM}$. This observation, based on the consideration that a large $\overline{GM}$ results in a so-called "stiff" ship, reflects the existence of a comparatively short and rapid period of roll and a quick recovery at the limits of roll.

In still water, a stable ship can be made to roll by application and subsequent removal of an external heeling moment. In the inclined position, a righting moment exists equal and opposite in direction to the heeling moment. When the external moment is removed from the ship in this position, the righting moment produces rotation of the ship towards the upright position. The potential energy existing in the inclined position (neglecting friction of the water) is entirely converted to energy of motion, so that in the upright position it is all kinetic energy. The ship will, therefore, continue its rotation to the other side until the kinetic energy has been reconverted to potential at the opposite limit of roll. Assuming no loss due to friction, the ship will continue to oscillate or roll indefinitely from side to side with constant amplitude. However, in practice, friction between the hull and water and radiating waves will dampen the oscillations as the original energy is gradually absorbed.

Subject to the condition that $\overline{GZ}$ is equal to $\overline{GM} \sin \phi$ for small angles, the period of roll will be independent of the amplitude ϕ . Because it is a difficult and tedious job to determine the value of k_x for any ship, it is possible to assume that k_x is a function of the beam and that it varies directly with the beam, and then fur-

ther state the equation 13-19 for roll empirically as

$$T = \frac{CB}{\sqrt{GM}} \quad (13-21)$$

where C is an empirical constant. (For large ships, C varies between 0.38 and 0.55, depending upon the ship and its loading.) B in this relationship is the extreme beam in feet.

A SHIP'S ROLL BY WAVE ACTION

The elements that cause a ship to roll in a sea are primarily the unbalanced moments resulting from a shifting center of buoyancy. As a wave approaches and passes under a ship, the waterplane is in a state of motion and is inclined at a rate depending upon the frequency, length, and amplitude of the wave. The center of buoyancy, whose position depends upon the slope of the waterplane at any given draft, will move out of a vertical line through the center of gravity if the waterline is inclined. The transverse component of this inclination results in an inclining arm being set up, whereby the ship will heel and tend to align itself so that the centerplane will be perpendicular to the wave surface. This action is, however, modified to some extent by the motion of the ship with the water itself.

Earlier on this chapter dealt with the way the water in a seaway involved in wave action moves in an orbital path. Halfway up the upper slope of a wave, the water particles are moving vertically upward; at the crest, they are moving horizontally with the wave front; halfway down the down slope, they are moving vertically downward; and in the trough, they are moving horizontally against the wave front. Thus, a ship floating in large waves will be influenced by the orbital motion of the water itself. At this time, there exists a centrifugal force that must be considered in addition to the forces of gravity and buoyancy. This centrifugal force is opposed by the dynamic force of the water that it produces, and consequently there are two distinct couples acting upon the ship and causing its rolling action: (1) the familiar one between buoyancy and gravity, and (2) the one between the centrifugal force of the ship moving in an orbital motion and the dynamic opposing force of the water. The periodic action of the first couple was described in the preceding section, where it was shown that the ship has a characteristic natural period of roll. The second couple is produced by forces that are entirely a function of the wave motion itself. The ship, and more specifically her center of gravity, moves in a circular orbit with a motion having the same period and orbital radius as the particles of water at the same level outside. The resulting centrifugal force, acting through the center of gravity and opposing the hydrodynamic force acting through the center of buoyancy, produces

the new couple that has the periodic motion of the waves in which the ship rolls.

Consequently, the resulting period of roll is a function of both the ship's natural roll period and the period of the waves. Such a statement, however, is incomplete without taking note of another fundamental fact. In the preceding section, a ship rolling in still water in its own natural period was subjected to only a single initial impulse and allowed to roll freely until the energy from this initial impulse was entirely consumed by the resistance of the water. This motion is known as *free oscillation*. A ship rolling in waves, however, is subject to nearly periodic impulses, with the result known as *forced oscillation* or *forced rolling* (see figure 13-14).

It should be pointed out here that these periodic impulses are the humps of an irregular sea, and that these waves tend toward regularity when they are the components of an underlying wave system.

Any suspended object, such as a pendulum, if free to oscillate when subjected to a series of forced impulses of regular frequency, will oscillate in the period of applied impulses. Thus the ship, if rolling in waves of exact uniformity and regular period, would eventually assume a period of roll identical to that of the waves. However, real sea waves are not constant in either period, amplitude, or wavelength and will, therefore, produce a series of more or less nonuniform impulses. Under such

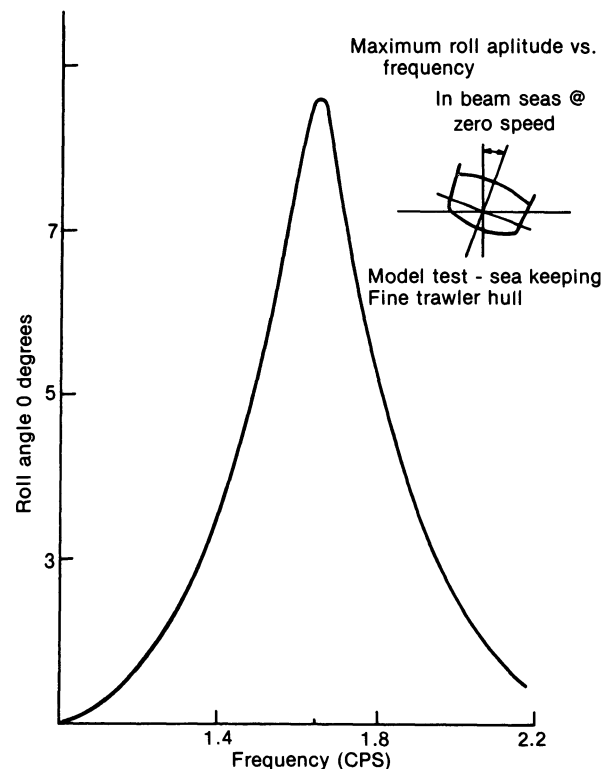


Figure 13-14. The roll amplitude related to the period for cyclic frequency in beam seas shows the sharp tuning phenomena of synchronism.

conditions, when the impulses producing the roll are not fairly regular, it is the ever-present tendency of the ship to revert to its own natural period of roll—a period that is the result of an independent impulse. The overall result is a period of roll for the ship that is a combination of her own natural period and the period of the waves producing the rolling moment, where the *latter* is generally the more predominant *period*.

SYNCHRONOUS ROLL

When the apparent period (that is, the interval at which the wave impulse strikes the ship) is the same or nearly the same as the natural period of the ship, a superposition of periodic inclining energies exists, and the result is an unduly heavy roll.

Such heavy rolling is not uncommon, nor is it difficult to distinguish from rolling or heeling caused by inadequate stability. However, synchronous rolling has frequently been mistaken and wrongly attributed to lack of stability. In most cases, nothing could be further from the facts. As will be seen immediately below, ships of large $\overline{GM}$ or large static righting moments are those that are more apt to encounter serious synchronous rolling. Ships of very low $\overline{GM}$ are much less frequently subject to such rolling.

It will be recalled from the equation for a ship's natural period of roll,

$$T = \frac{CB}{\sqrt{\overline{GM}}}$$

that the period varies inversely as the square root of the metacentric height. Therefore, the greater the metacentric height for the same beam, the shorter is the natural period of roll. At the same time, for larger vessels, the shorter the period of roll (12 seconds and below), the greater is the probability of synchronism among large sea waves. For example, large Atlantic storm waves are about 500 to 600 feet between crests (wave length) and have a period of 10 to 11 seconds. Under such conditions, a large ship of moderate or low metacentric height would have a period in excess of the period of these waves and would be comparatively steady in such a seaway. On the other hand, a similar ship of fairly large $\overline{GM}$, with a period of approximately 10 to 11 seconds, would be an extremely bad roller in this situation.

Furthermore, the factors determining the period of roll are the metacentric height and the beam as a function of the radius of gyration. It may be noted that approximate typical periods of a carrier and a destroyer in normal conditions are 17.0 and 8.0 seconds respectively. This would indicate that the destroyer of smaller $\overline{GM}$ might increase her natural period to be in synchronism with a heavy sea with a *decrease* in metacentric

height. This is a more likely situation than that where the carrier, in the case cited, would have to increase her $\overline{GM}$ to approximately 16 feet before the period would decrease to a synchronous value. Therefore, with the existing variety of ship sizes, shapes, and consequent variable values of $\overline{GM}$, it would be wiser to analyze any situation as a separate problem, noting only in general that for most large oceangoing vessels, a relatively high $\overline{GM}$ may result in a period-producing synchronism in a heavy sea.

It should be noted also that naval vessels are designed with metacentric heights dictated by the requirements of stability in probable damaged conditions, beam winds, high-speed turning, etc. The metacentric heights of such ships are therefore maintained at relatively high values. Oceangoing passenger vessels have a much lower range of values of $\overline{GM}$, and thereby fulfill the requirement of comfort.

13-8 Pitching and Heaving

Pitching may be considered as a phenomenon analogous to rolling, except that the axis of rotation is at 90° to the rolling axis in the same plane. The undamped natural period of pitch is usually between one-third and two-thirds of the natural period of roll, as estimated from equations 13-19 and 13-20.

Where pitch, yaw, and heave are concerned, it is more difficult to describe ship motion as a single phenomenon, as can be done in the case of roll. Particularly interrelated are pitch and heave, which are together influenced by roll, yaw, sway, and surge. To describe these predominant motions mathematically requires certain linearized theories, such as "strip" theory, together with the development of coupled differential equations. The entire theoretical study of ship motion is introduced in the more advanced courses of naval architecture.

Pitch-heave motion, acting like it does in a real sea as a coupled motion, produces several objectionable conditions of ship operation, the most undesirable of which are reduction of speed, slamming, and wet decks, and the related interference with human functions and machinery.

Apparently, moderate variation in hull form contributes relatively little to the reduction of synchronous pitch phenomena. There are certain damping factors, however, that can partially eliminate or noticeably reduce some of the objectionable conditions. In hull-form design, a sharp or fine entrance combined with hollow, flaring sections produce an early and easily excited pitching motion. This condition may be offset by more convexity in the forward and after sections. There is considerable interest in seeing if optimum hull forms can be developed that combine good seakeeping characteristics with good fuel economy at cruising speeds.

These criteria are frequently in conflict for certain ship types, so the optimization process is a compromise based on relative weights given the various factors.

13-9 Yawing

Yawing is the angular motion about a vertical axis through the ship's center of gravity. The motion may be generally said to be the result of three factors other than poor steering: (1) inequality of static pressures on the hull, (2) orbital motion of the water in a seaway, and (3) gyroscopic action.

Generally, in a seaway, the wave profile on the port and starboard sides of the ship is not the same, and as a result, the longitudinal position of the center of pressure on one side of the submerged portion of the ship is offset both longitudinally and vertically from that on the other side. This produces a rotating couple about the vertical axis, or a yawing tendency, as well as a heeling moment. As the wave profile changes with the passing seas, the yawing couple changes in magnitude and alternates in direction, producing an oscillation. This oscillation will occur in the apparent period of the waves passing the ship. It is best corrected by anticipating the motion and meeting it with compensating rudder action.

In addition to the aforementioned quasi-static forces producing yawing, a dynamic yawing action is produced by the orbital rotation of the water in a wave. It will be recalled that the particles at the crest of the wave are moving at the top of a circular orbit in the direction of wave advance; in the trough of the wave, they are at the bottom of their orbit and moving in a direction opposite to the wave advance. Thus a ship moving in a quartering sea or with the sea at an angle on the bow is subjected to a yawing couple (see figure 13-15).

As the wave passes the ship, changing from the crest to the trough at the bow and from the trough to the crest in the after portions of the ship, this couple is reversed. The net result is a yawing oscillation with the same period as the period of encounter of the waves.

Rudder compensation for yaw from this dynamic and orbital motion of water is more difficult. This is because, every half wavelength, the water in the vicinity of the rudder will be moving in the same direction as the ship, and a severely reduced turning couple can be developed by the rudder. This loss of control is especially critical if the rudder is not located in the propeller race. If the seas are large and this dynamic yaw develops to serious proportions, the best means by which to combat it on a multiple-screw ship is the opposed variation of pro-

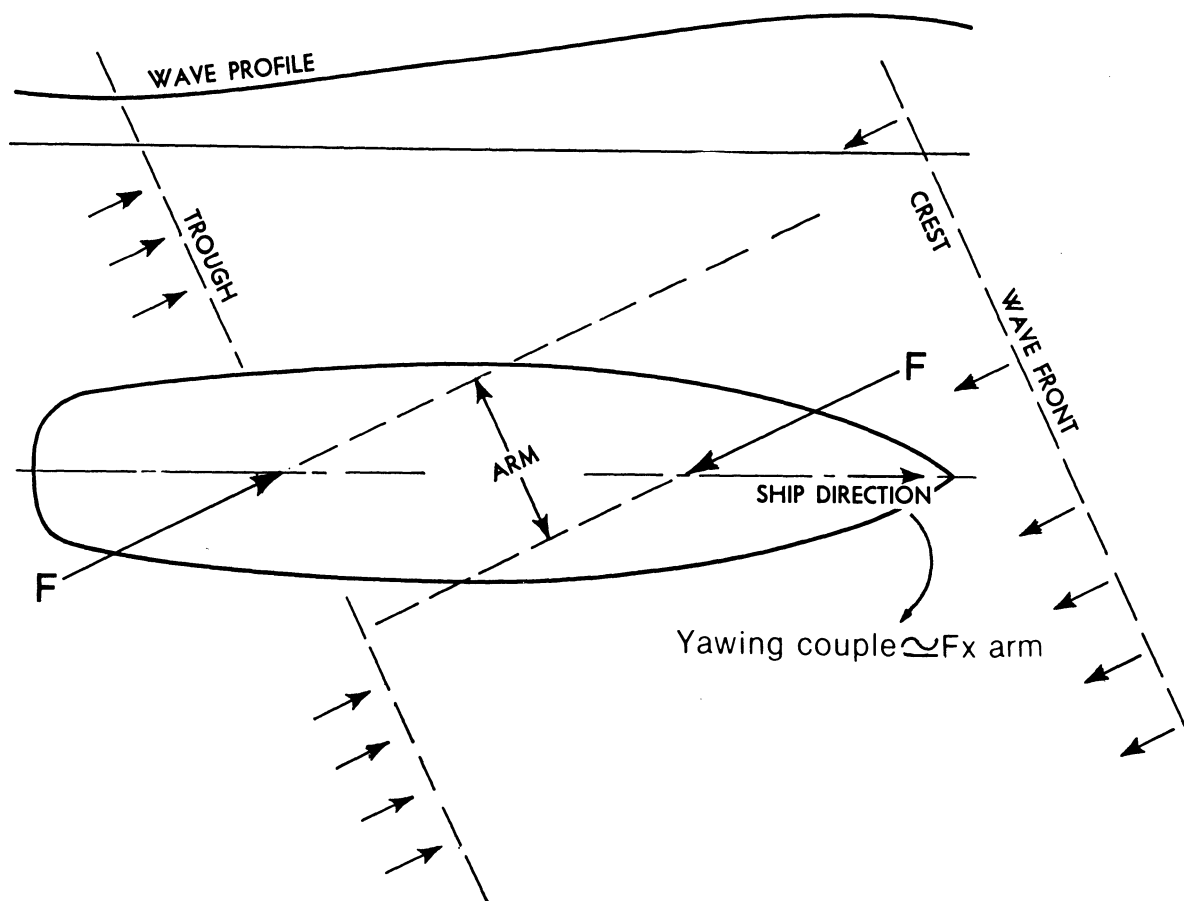


Figure 13-15. Yawing couple in a seaway

pellor speed. In some cases, loading to trim by the stern may be helpful.

13-10 Motion-Damping Devices

It has been the ship designer's unhappy lot through the centuries to cope with the difficult movement of the ship in a seaway and to try to devise means to damp the more objectionable of ship motions, primarily roll and pitch. The reduction of these motions as well as yaw has been the object of extensive research and many odd inventions. Most of the many devices of the past, as, for example, the passengers' salon of an early nineteenth-century English Channel ferryboat that was suspended in gimbals, have been for one reason or another only partially successful. There have been in recent years several notable exceptions, and successful reduction of excessive rolling to almost negligible proportions has been (where economically justified) accomplished. The mechanical problems have not been entirely solved, nor indeed has any single device emerged more successful than any of the others.

In general, all stabilization systems depend on the motion of mass and may be classified properly as follows:

1. Type of force utilized
 - a) counterweight—gravitational force
 - b) acceleration—inertial force
2. Location of system
 - a) internal
 - b) external
3. Type of mass
 - a) solid
 - b) liquid

Not all of the above types will be described as examples of applications here—only those that are most successful and more frequently used. In the case of antiroll devices, these include

1. Bilge keels
2. Controllable fins
3. Antirolling tanks
4. Gyrostabilizer (active type)

(Cox and Lloyd 1977).

BILGE KEELS

The long fin-like projections attached to most ships along the turn of the bilge and extending from one-half to two-thirds of the ship's length are normally referred to as *bilge keels* or *antirolling keels*. These attachments are undoubtedly the simplest, probably the oldest, and one of the most successful and economical means of reducing the rolling motion of a ship.

The bilge keel is generally a continuous attachment of a single, heavy steel-plate structure on large ships that projects from 2 to 4 feet from the hull and is ap-

proximately normal (perpendicular) to the hull surface. On larger ships, the bilge keel is of a V-shaped cross section and is generally filled solidly to prevent crushing when docking or grounding (see figure 13-16).

On some smaller craft, the bilge keel may be of light steel plate and attached by welding at regular short intervals and undercutting in between the welded attachments.

Regardless of the specific structural shape and details, bilge keels operate according to a very simple and effective theory. Recalling the equation

$$T = \frac{1.108 k}{\sqrt{GM}}$$

it was pointed out that k was the radius of mass gyration of the ship in roll. With bilge keels projecting from the sides of the ship causing a somewhat increased mass of water to roll with the ship, the value of k in the above equation is increased, thus increasing the period of roll. Under forced rolling conditions encountered in a seaway, with the increased natural period the amplitude of roll is decreased. The major effect of bilge keels, however, is the increased resistance to roll due to the viscous-eddy effect.

From both test results and observed results, it can be seen that bilge keels are more effective on a ship when moving ahead through waves than when stopped. Such observations suggest that there is a hydrodynamic lift in the forward sections of the bilge keels. This hydrodynamic lift resists the lateral forces of roll and adds to the steadiness of the ship. In this sense, bilge keels are essentially elongated fins acting as hydrofoils and are a special application of fixed stabilizing fins. Fins and hydrofoils will be more fully explored in subsequent sections, but in relation to bilge keels, further tests have shown that their dynamic effect could be improved by constructing them in shorter sections rather than in the conventional single, continuous section commonly used. However, where dynamically suppressed roll is desirable, it is more effective to use controlled stabilizing

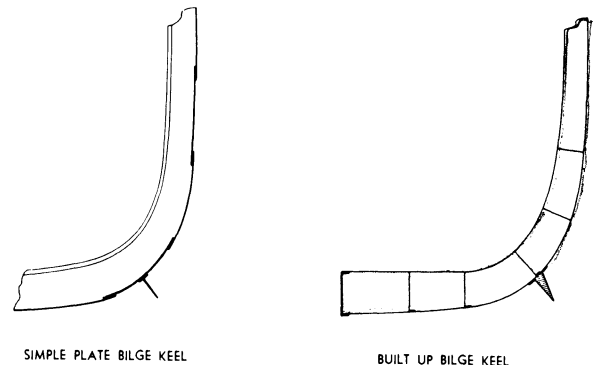


Figure 13-16. Bilge keels

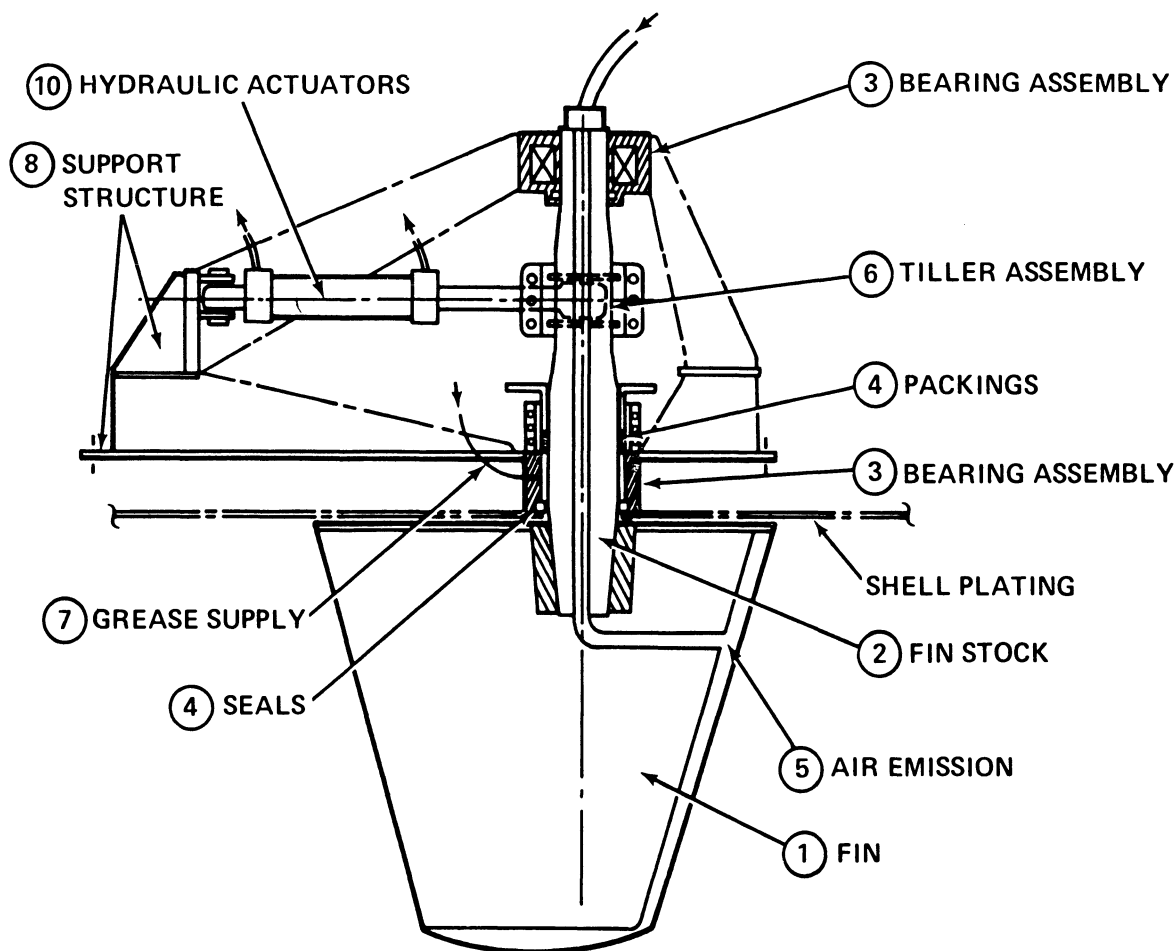


Figure 13-17. Schematic of fin and hydraulic subsystems. (From Nelson and McCallum 1978.)

fin. It should be pointed out that bilge keels must be considered as a means of reducing roll with no attempt toward nearly complete elimination. There must be a rolling action of some magnitude for them to be effective. They are roll-damping devices of a static type, requiring no energy supply.

That bilge keels are used on almost every ship regardless of size or type attests to their adaptability. Their simplicity, economy, and effectiveness make them universally attractive. Their principal disadvantage is the added resistance to forward motion that occurs during rolling and during off-design speed operation, since the flow streamlines may no longer be in line with the bilge keels.

CONTROLLABLE FINS

Stabilizing fins that could be controlled were patented in 1889 by John Thornycroft in England but were not actually applied until about twenty-five years later by Motora in Japan. Since World War II, there has been an ever-growing application of controllable fins, and large ships and pleasure craft have been built with or have had the devices installed in them. Credit for the

successful current development of this type of device must be shared by the Denny-Brown Company of the United Kingdom and the Sperry Gyroscope Company of this country.

Briefly, the device consists essentially of a projecting fin (actually one on each side of the ship) at the bilge line and somewhat forward of amidships (see figure 13-17). Some fins are retractable either axially or radially and, when fully extended, can rotate within a limited arc in a similar manner to a stabilizing fin on an aircraft or the diving planes on a submarine. In other fins, tail flaps are fitted to the fin's after edge to produce a shaped hydrofoil section that will produce the most effective vertical force (either upward or downward) when the angle of attack is reversed. The angle of attack of these fins is controlled through a shaft connected to motors inside the ship. A gyroscopic sensing device actuates the motors, creating a response to and, in fact, anticipating the wave's roll force (Nelson and McCallum 1978). The transmission of motion to the fins produces at the proper time the desired angle and results in a force at the fins opposing the heeling or rolling force of the waves. The port and starboard fins operate simultaneously with a 180° phase relationship, thus pro-

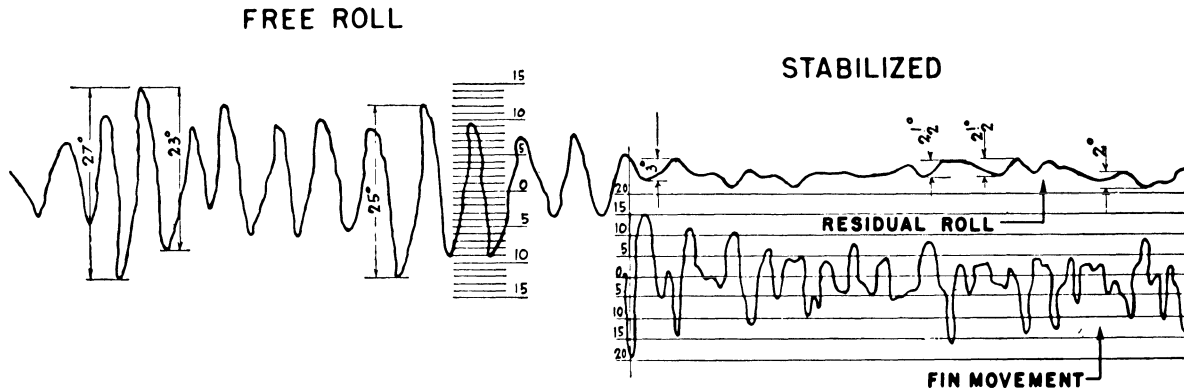


Figure 13-18. Actual rolling record with and without fins operating

ducing a total correct moment opposite in direction to the rolling moment. Figure 13-18 shows a typical comparison under given sea conditions of both unresisted rolling and rolling under control of a fin stabilizer of the type described above.

ANTIROLLING TANKS

There have been many types of internal tanks designed to produce counter-rolling moments, and they have been tried with varied degrees of success. No attempt will be made here to describe all these systems. Rather, it will be more pertinent to single out one of the most successful of antirolling devices.

The *Frahm* antirolling tank consists basically of a U-shaped tank system transversely arranged from side to side. Actually, the system contains two vertical legs (one to port and one to starboard) connected by a horizontal leg. The horizontal leg is of a smaller sectional area than the vertical legs, and when the system is approximately half filled with water, it is so designed that the natural period of oscillation of the water is approximately equal to that of the ship—actually slightly less. The vertical legs are connected across the top by an air line whose dimensions are also critical in affecting the period of the water in the system. This air line is equipped with valves that, upon adjustment, can control the amount of water transferred between legs and the phase relation between the roll of the ship and the transfer of the water. The phase lag is adjusted to about 90° so that the water in the horizontal leg is always running downhill to the low tank, creating a damping moment and resisting the roll of the ship.

It is customary to locate the tanks above the center of gravity of the ship. This is primarily because the moment of force due to the horizontal acceleration of the water should be in the same direction as the static moment of the water in the vertical legs.

Figure 13-19 is a schematic sectional sketch showing the arrangement of *Frahm* tanks, with a record of the

ship's oscillation with the system, not operating and then operating.

GYROSTABILIZERS

The action of a gyrostabilizer is based on the elementary theory of a gyroscope. Assume a gyroscope is spinning in its frame with its axis XX' vertical, as in figure 13-20. The frame is attached through bearings to the structure of the ship along axis YY' . Now suppose the ship is subjected to a rolling moment, that is, motion about its longitudinal axis. This will result in a precession of the gyroscope's axis and a resisting couple set up to the initial rolling moment. Such a simple arrangement as described above involves the basic gyro, which could be classified as the passive or responsive type. However, the Sperry Gyroscope Company has improved it, and it is now classified as an active type. As applied in ships, it is a pilot gyro. The pilot gyro actuates the primary gyro, causing it to precess in a direction opposite to that which would result from the rolling of the ship. Gyrostabilizers of this type have been used successfully for many years, particularly in smaller ships, such as pilot vessels and yachts. The largest vessel in which the gyro system has been used was the passenger liner of the pre-World War II period, the *Conte di Savoia*, an Italian luxury liner of more than 40,000 tons displacement. This ship contained three gyrostabilizers with rotors 13 feet in diameter, whose aggregate weight when installed was 690 tons, or about 1.72 percent of the displacement. In operation, this installation accomplished about 60 percent stabilization (reduction of roll). While this is a fair amount of stabilization, it by no means represents the best performances of gyrostabilizers, which have shown the ability to reduce roll up to 80 percent.

Gyrostabilizers are better installed on small- or moderate-sized ships because of the structural limitations of size and mass. Larger vessels today depend almost ex-

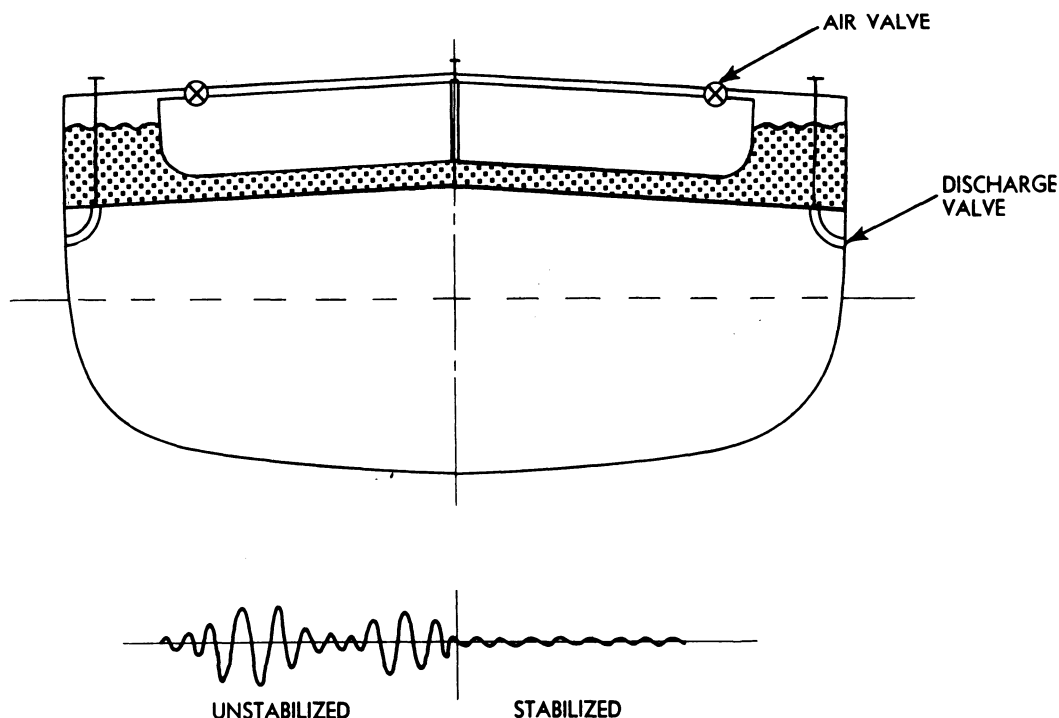


Figure 13-19. Frahm antirolling tank with typical stabilizing record

clusively on controllable fins or antiroll tanks for effective roll stabilization.

Problems

13-1. A destroyer is advancing at 20 knots in beam swell waves. These waves have an approximate wavelength of 300 feet. Is the frequency of encounter of these waves close to the natural roll frequency of the ship?

Assume

roll radius of gyration $k_x = 0.64 (B/2)$
 added-mass coefficient in roll $x_A = 0.25$
 $\overline{KG} = 15.0$ feet
 $\overline{KM} = 19.4$ feet
 $\Delta = 3,100$ tons
 beam $B = 40$ feet

length $L = 383$ feet
 draft $T = 13$ feet

13-2. Consider an FF 1052 with a mean draft of 15.5 feet, a $\overline{KG}$ of 16.5 feet, a pitch radius of gyrations of $0.25 L$, a roll radius of gyration of 0.32 times beam, and an added-mass coefficient for pitch and heave of 1.4 . From the curves of form,

added-mass coefficient in roll $x_A = 0.25$

$L_{PP} = 415$ feet

$B = 45.75$ feet

displacement = 4,250 tons

$\overline{KM} = 22$ feet

$\overline{TPI} = 33.5$ tons/in

$\overline{BM}_L = 960$ feet

$\overline{KB} = 9$ feet

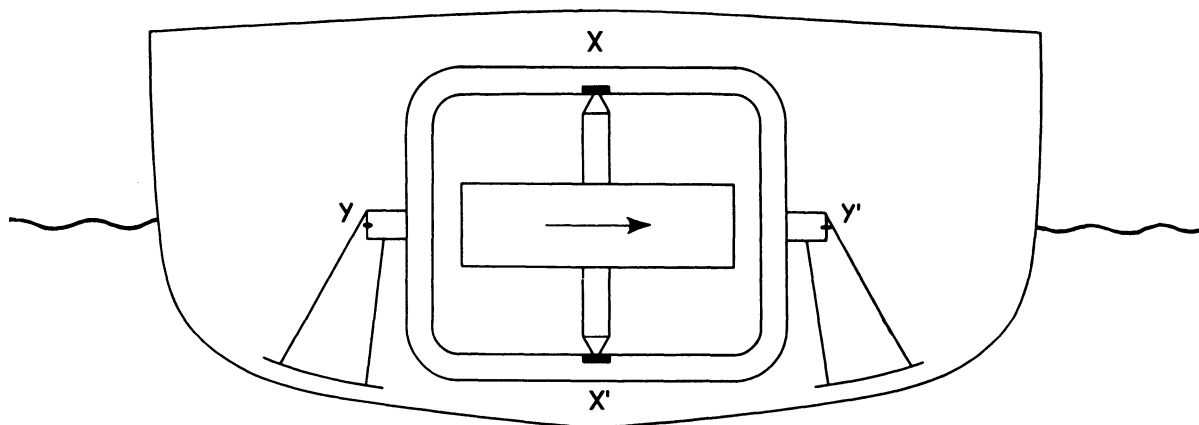


Figure 13-20. Simple gyroscope stabilization

- a) Determine the natural periods in pitch, heave, and roll.
- b) Determine the critical wavelength for synchronous rolling in beam seas.
- c) Determine the critical speeds for synchronous pitching and heaving in head seas for wavelength-to-shiplength ratios of 1.0, 1.25, and 1.5. For head seas, determine the pe-

riod of encounter if found from

$$T_{\text{encounter}} = \frac{L_{\text{wave}}}{v_{\text{ship}} + c_{\text{wave}}}$$

13-3. Write a computer program to calculate critical speeds and wavelengths of various ships based on the parameters given in problems 11-2 and 13-2.

Maneuverability and Ship Control

14-1 The Ship's Rudder

The rudder is the most important ship control surface and certainly the oldest. This vital device, or its equivalent at the ship's stern, has been in use since prehistory. In the earliest archeological pictures of man's watercraft, there are very prominent steering oars with wide, rudder-like blades held to the stern.

In a technical sense, a great deal of improvement in the rudder as an effective control surface can be seen; but very little change in its basic nature and usage has taken place.

Historically, there have been only about three fundamental design changes in this steering appendage. The first change took place sometime during the thirteenth century, when the long, heavy steering oar, swiveled on the side near the stern (it had been so effective, apparently, as to dissuade mariners from making any real change for nearly five thousand years) was replaced by a hinged, centerline blade hung on the stern post. Not until the era of iron ships in the latter part of the nineteenth century did the rudder ever penetrate the hull below the waterline, and only in very recent decades has the rudder begun to be dissociated from the last vestiges of a stern post and become a truly separate control surface. There have been only a few exceptions to the above historical trend.

In its present, most adaptable form on large and even small ships, the rudder is a fin- or spade-like projection under the counter and below the waterline, generally placed as far aft as practicable. It is hung on a circular, solid shaft, called a *stock*, that penetrates the hull through a stuffing box and bearings. It often has a fixed, faired, foil-like section ahead of it, which is firmly attached to and part of the ship's structure. This is called the rudder *horn* in the case of a separate faired sternpost. The fixed structure may also be part of the skeg or *deadwood*. In any case, it serves a double purpose: to help support

the weight and thrust of the rudder and to improve materially the hydrodynamic effectiveness of the rudder. The basic dimensions and nomenclature for an *all-movable* rudder without horn or skeg are shown in figure 14-1.

14-2 Control Surface Definitions

The basic definitions used to describe the shape and hydrodynamic characteristics of rudders and other ship control surfaces (foils) are the same as those used to describe the various lifting surfaces (airfoils) in aerodynamics.

1. *The mean span ($\bar{b}$)* is the average of the spans of the leading and trailing edges of the control surface (figure 14-1).
2. *The mean chord ($\bar{c}$)* is the average fore and aft distance between the leading edge and the trailing edge, or the average of the *root chord* and the *tip chord*, as shown in figure 14-1.
3. *The profile area (A_p)* is the projected area (or planform area) of the control surface and may be taken as the product of the mean span and the mean chord.
4. *The aspect ratio (AR)* is the ratio of the mean span to the mean chord $\bar{b}/\bar{c}$, or the ratio of the mean span squared divided by the profile area $\bar{b}^2/A_p$.
5. *The taper ratio* is the ratio of the tip chord to the root chord.
6. *The sweepback angle* is the angle between the quarter chord line and a line perpendicular to the centerline of the ship.
7. *The mean thickness (t)* is the average of the maximum thickness of the foil at the root and the tip.
8. *The fineness ratio* is the ratio of the mean thickness to the mean chord.
9. *The angle of attack (α)* is the angle between the mean chord line and the direction of the *free stream velocity (U)* as shown in figure 14-2.

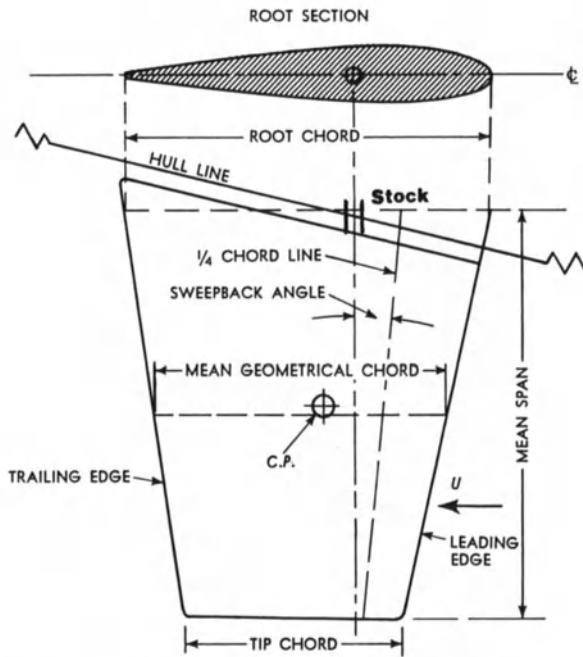


Figure 14-1. Basic rudder definitions for spade rudder

10. *Lift* (L) is the component of the resultant force on a control surface (foil) that is perpendicular to the direction of motion (free-stream velocity). It is calculated from

$$L = C_L \frac{1}{2} \rho v^2 A_p \quad (14-1)$$

11. *Drag* (D) is the component of the resultant force on the control surface (foil) parallel to the direction of motion. It is calculated from

$$D = C_D \frac{1}{2} \rho v^2 A_p \quad (14-2)$$

12. *Normal force* (F) is the component of the resultant force on a control surface that is perpendicular to the longitudinal axis of the ship.

13. *The center of pressure* ($C.P.$) is the point on the foil through which the resultant force may be considered to act. It is located at the quarter chord position for thin, symmetrical foils of infinite aspect ratio but varies with angle of attack for low aspect ratio foils.

14. *The camber* of a foil is the maximum distance between the chord line and the mean line, which lies halfway between the outer surfaces of a foil. (A cambered foil can generate lift at zero angle of attack since it is unsymmetrical with the flow at this condition.) Since rudders must be equally effective to port and starboard, they are generally symmetrical about the chord line and therefore have zero camber.

15. *The rudder torque* (Q_R) is the torque applied to the stock or actuating mechanism of a rudder by the hydrodynamic forces acting upon it.

14-3 Area and Shape of Rudders

There is no fixed rule for the determination of the size of rudders owing to the difference in maneuverability required for different ships. In practice, rudder area, expressed as a fraction of the product of the length and draft or centerline plane area, is often selected by comparison with another similar ship that has the required maneuverability. The ratio of rudder area to the product of length and draft for destroyers is usually about .025, while for a cargo ship the ratio is usually about .017.

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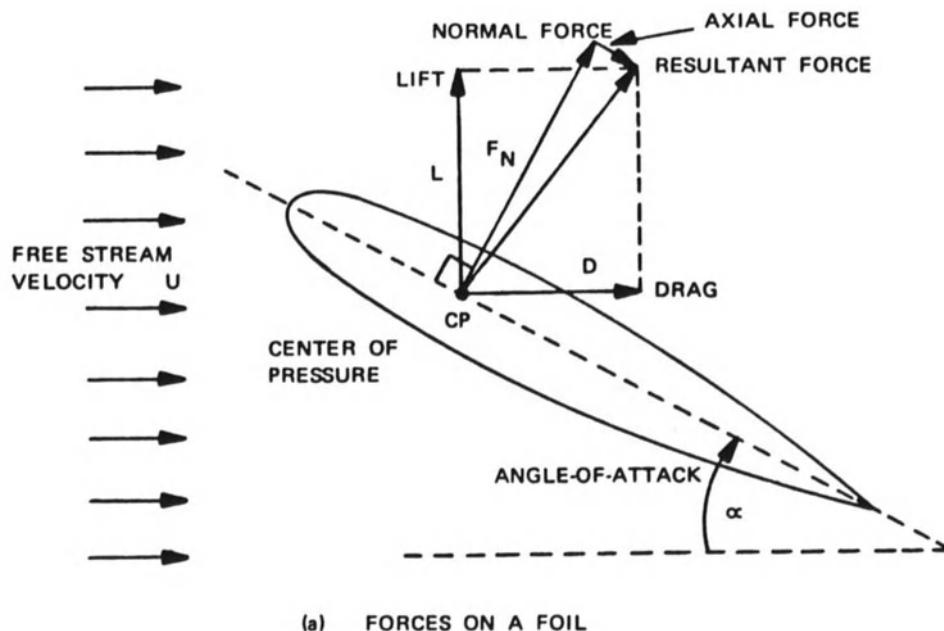


Figure 14-2. Rudder force nomenclature

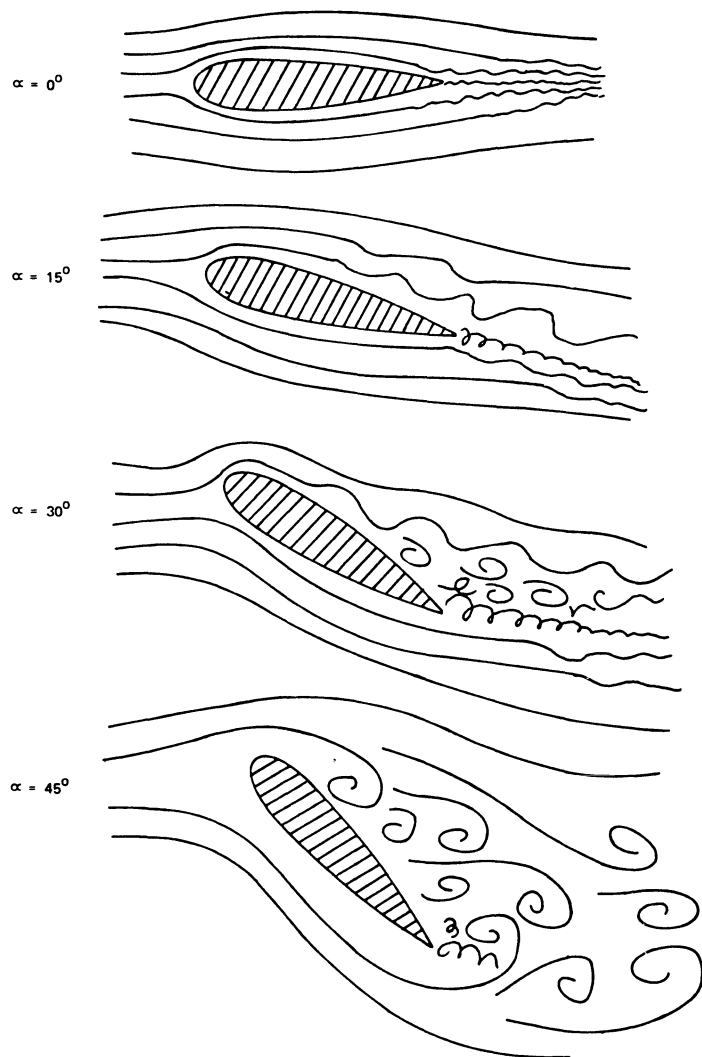


Figure 14-3. Progression to a stall

Veritas suggested the following minimum area for rudders working directly behind a propeller:

$$A_P = \frac{LT}{100} \left[1 + 25 \left(\frac{B}{L} \right)^2 \right]$$

where L , B , and T refer to length, beam, and draft of the ship.

Rudders can also be sized from model maneuvering tests (to be discussed later).

The most significant shape parameter for control surfaces (foils) is the aspect ratio or ratio of mean span to mean chord of the movable portion of the surface. The flow around very high aspect ratio foils is nearly two dimensional over much of the foil. The effects of the tip vortex generated by the difference in pressures on either side of the foil near the tip (similar to those visualized in propeller-tip cavitation in figure 12-6) are confined to the outer portion of a high aspect ratio foil, but they can significantly affect much of the flow over a low aspect ratio foil. The effect of the tip vortex is to reduce the apparent angle of attack and to cause what

is known as *induced drag*, that is, drag induced by lift. Since induced drag reduces the *lift-to-drag ratio* at a given angle of attack, the highest lift-to-drag ratios are found on high aspect ratio wings and foils. (This is why gliders and soaring birds have high ratios of mean span to mean chord.) High aspect ratio control surfaces have higher lift coefficients for a given angle of attack (or rudder angle), but they also *stall* (induce flow separation on the low pressure side of the foil) more at lower angles of attack than low aspect ratio foils. When a foil stalls ("bubbles"), the lift drops sharply and the drag increases rapidly, reducing the effectiveness of control surface (figures 14-3 and 14-4).

Most ship rudders can be rotated up to 35° port or starboard, and they tend to have aspect ratios less than 2. Some spade rudders on sailboats (supported only at the hull line) have higher aspect ratios, since the potential for grounding damage is reduced by the extended keel. Protection from grounding damage tends to limit the span of rudders to less than that which would extend below the keel line.

The effective aspect ratio of a rudder can also be increased at low rudder angles by minimizing the gap between the inboard side of the rudder and the hull line. If this space is small enough in low deadrise ships, formation of a strong tip vortex is inhibited (as it is near the tip of a ducted propeller). This increases the effective angle of attack near the root chord and yields lift-to-drag ratios equivalent to a higher aspect ratio foil. It also increases the possibility of flow separation (stall) at lower rudder angles.

The rudder consists of two parts: the flat part or blade, against which the water pressure acts, and the shaft or stock, which transmits the motion of the steering gear to the blade. In the case of warships, a principal concern is to get sufficient blade area totally below the waterline, with the bottom and trailing edge above the base line and forward of the tangent to the after-contour of the ship.

The general types of rudders are unbalanced, semi-balanced, or balanced, as shown in figure 14-5. The blade of an *unbalanced rudder* is entirely aft of the stock. In a *balanced rudder*, a portion of the rudder area, disposed symmetrically throughout the rudder height, is forward of the stock. In a *semibalanced* rudder, the area forward of the stock does not extend to the full height of the blade aft of the stock. Hence, the upper portion of a semibalanced rudder may be considered an unbalanced or hinged rudder, and the lower portion a balanced rudder.

Disposing a portion of the blade area forward of the stock reduces the torque. Since the center of pressure moves toward the trailing edge as the rudder angle is increased, it is not possible for the stock and center of pressure to coincide at all rudder angles. Balanced rudd-

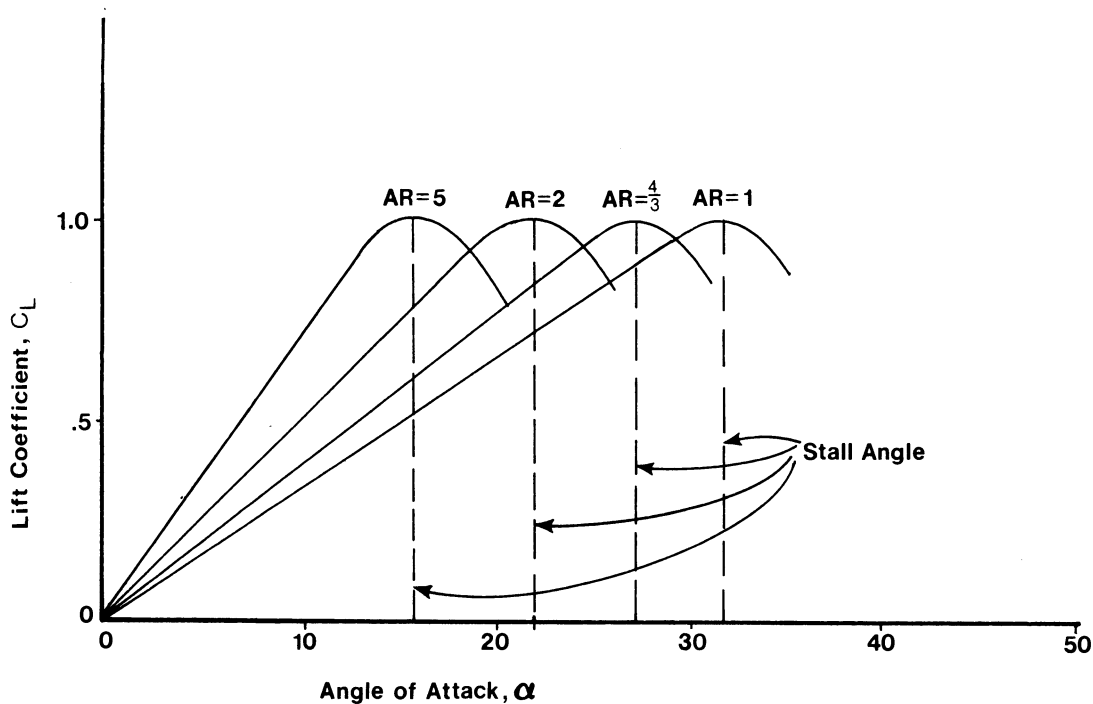


Figure 14-4. Effect of aspect ratio on stall angle

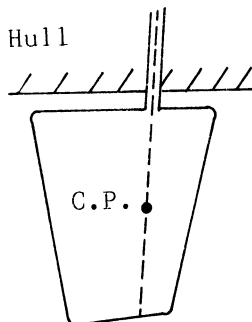
ders are normally designed to be balanced (that is, to have the center of pressure coincide with the rudder stock) at about 15° of rudder angle.

Figure 14-6 illustrates the relationship of force, location of center of pressure, and torque on a balanced rudder. It will be noticed that the location of the center of pressure is a function of the rudder angle and moves aft as the rudder angle is increased, being on the axis of the rudder at about 15° . The product of the force on the rudder and the distance to the center of pressure from the axis produces the torque variation shown in

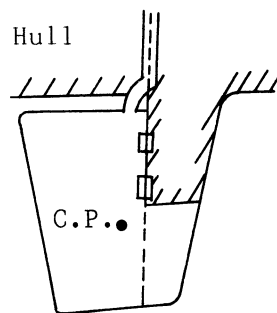
curve 3 of figure 14-6. This curve actually represents the mean torque on the rudder and not the actual torque required by the steering engines. Because of friction, the actual amount of torque required by the steering engine varies not only with the angle but with the increasing and decreasing angle. The actual torque supplied is indicated by the dotted curves above and below curve 3.

A streamlined double-plate rudder develops more rudder force and has less resistance than a single-plate rudder. Considerable attention to the design of the rudder

All movable
Balanced
Spade
Rudder



Semi-balanced
Rudder on
a horn



Unbalanced
Flapped
Rudder on
a Skeg

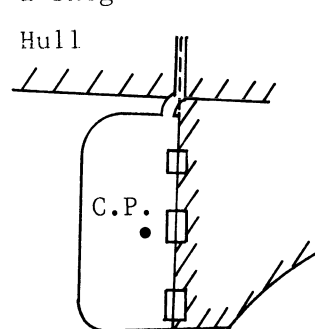


Figure 14-5. Rudder types

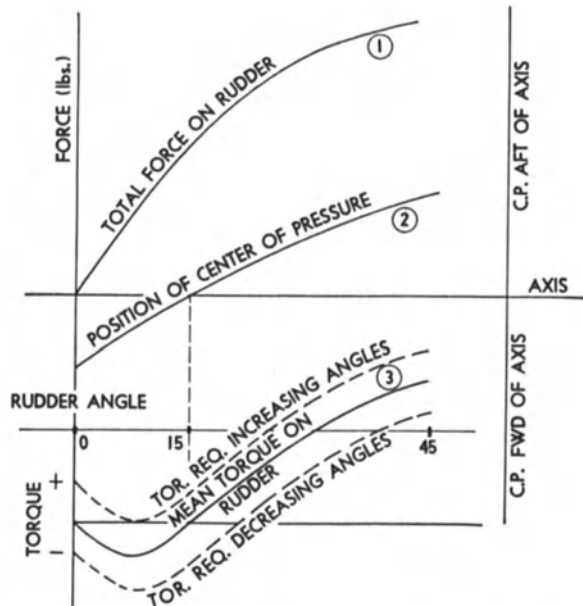


Figure 14-6. Rudder angle versus force, center of pressure, and torque

der post and rudder will materially reduce resistance (see figure 14-7). Modern destroyers are fitted with twin spade rudders to avoid the space limitations of a single rudder and to enjoy the greater effectiveness achieved when each one is located in the race of each of the twin screws. Frigates generally have a single spade rudder located in the race of a single screw.

FLOW AROUND RUDDERS

The flow phenomena and resultant forces on the rudder are generally treated with the same theories as those used in basic aerodynamics. There are, however, some marked differences and special considerations that combine to create a more complex and variable situation for a rudder performing its function for a ship. The rudder operates in a complicated environment—in the vicinity of the stern, aft of the propellers and generally either wholly or partially in their race, and often in the zone of separation, where strong eddy currents exist and where ship motion and sea state produce oscillating forces and velocities. Basically, there are three hydro-

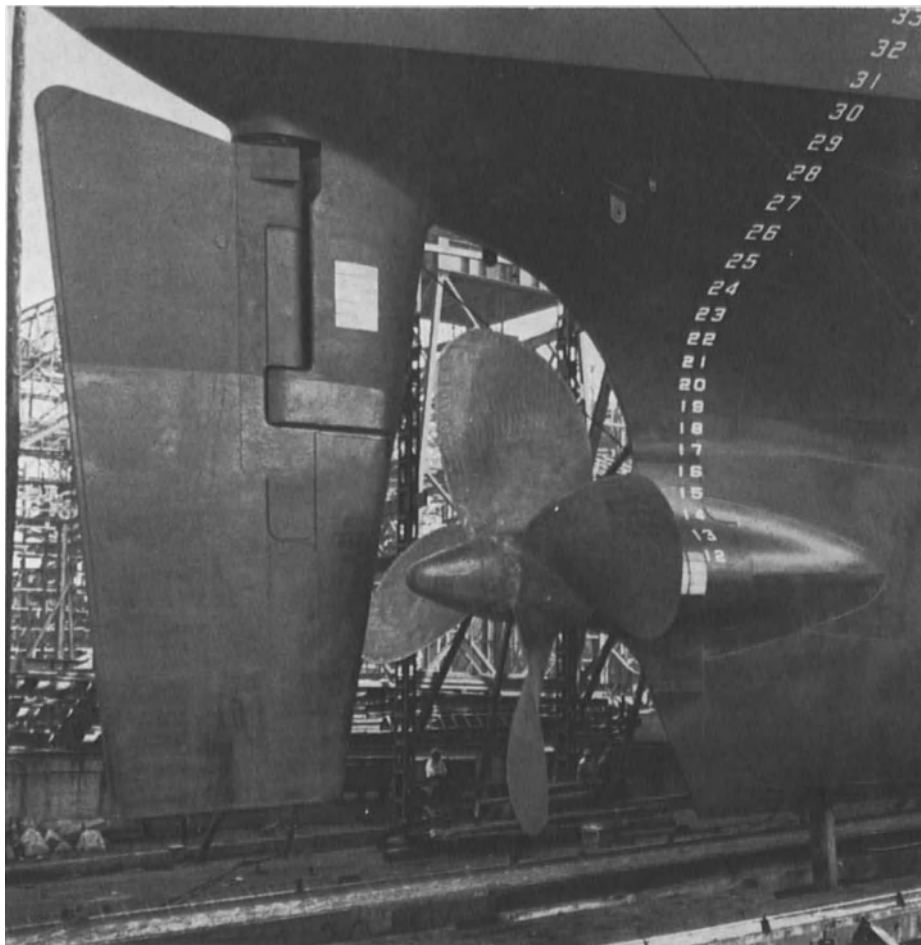


Figure 14-7. Typical and efficient modern, double-plate, semi-balanced rudder (on horn) in a single-screw ship. (Courtesy of Newport News Shipbuilding.)

dynamic flow phenomena to which a rudder is subject, even when it is operating in a free stream apart from interaction with the ship: *hydrodynamic stall*, *cavitation*, and *aeration*. These phenomena may exist separately or together. They all inhibit or, occasionally, destroy the effectiveness of rudder action.

Stall is the same phenomenon encountered on an aerodynamic surface. It is the sudden discontinuity of lift on the downstream surface of the rudder caused by increasing the angle of attack to the critical angle, where separation occurs when the normal flow pressures can no longer exist.

Cavitation and *aeration* are functions of the magnitude of pressure reduction on the downstream side of the rudder. Aeration, or ventilation, often results when the rudder is too close to the surface, either by design or motion of the sea, and air is pulled down from the atmosphere to the low-pressure region of the rudder. Cavitation is the same phenomenon as that produced under certain conditions on propeller blades and hydrofoil surfaces, which is due to increasing reductions in pressure leading to the vaporization point. (It is discussed in chapter 12 in more detail.) It not only inhibits lift or rudder force but produces erosion on the metal surfaces in its vicinity.

These hydrodynamic flow phenomena, where the rudder is actually operating in its real environment on the ship, are hard to predict. Scaling factors in model tests become difficult to determine because all of the above phenomena are governed by different laws of similitude as well as influenced by flow velocities induced by the turning ship. Model tests are helpful in qualitatively determining rudder effectiveness, and free-

stream rudder experiments in flow channels at high Reynolds numbers and in turbulent flow are often reliable, particularly where the parent-hull flow conditions have also been investigated.

14-4 Factors Affecting Steering

The portion of a ship's hull aft in the vicinity of the keel and just forward of the propeller is in the form of a vertical fin-like surface apparent in the sharp V-shaped sections. This fin-like surface, called the *deadwood*, is not the same either in extent or shape from ship to ship. It may be extensive and predominant in some types of ships, particularly single-screw, oceangoing vessels, and it may be extremely limited in other types. Its purpose, in any event, is to provide the desired degree of directional stability. The turning resistance is greatly increased by a large amount of deadwood or skeg area, and the ships with a small amount of deadwood tend to yaw in a heavy sea. In large ships like aircraft carriers, where directional stability is a critical factor, it is often necessary to provide twin skegs, spaced side by side on each side of the centerline (see figure 14-8). In ships such as destroyers, where high-speed maneuverability is desirable, a proportionately smaller amount of deadwood is used.

Ships with large moments of inertia about their pitch and yaw axes are less sensitive to small sea disturbances and will steer better in moderate seas.

The effectiveness of the rudder and the time required to apply the rudder contribute to steering characteristics. To avoid building up a large angular momentum due to a disturbing force, the helmsman must detect and oppose the initial tendency to yaw.

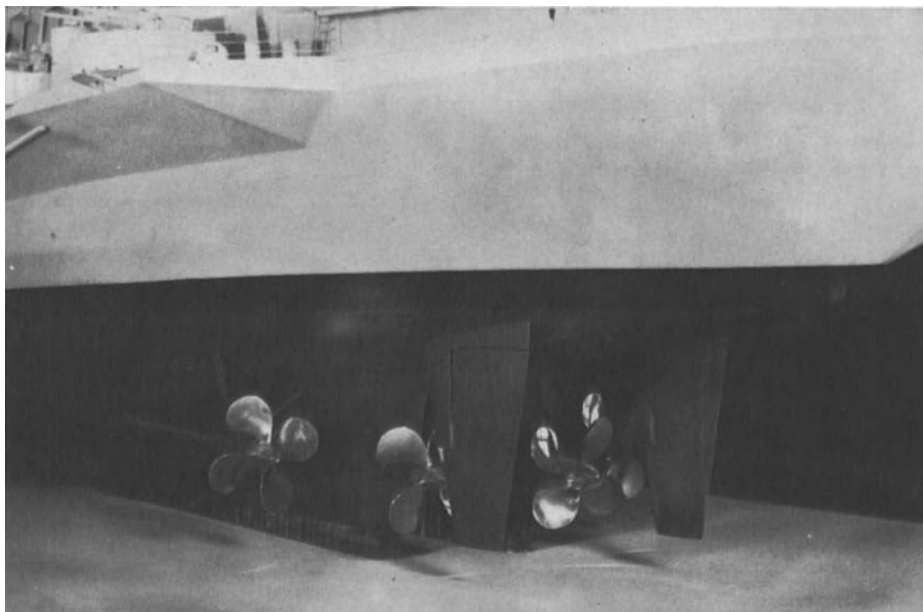


Figure 14-8. Twin skegs and rudders on large carrier

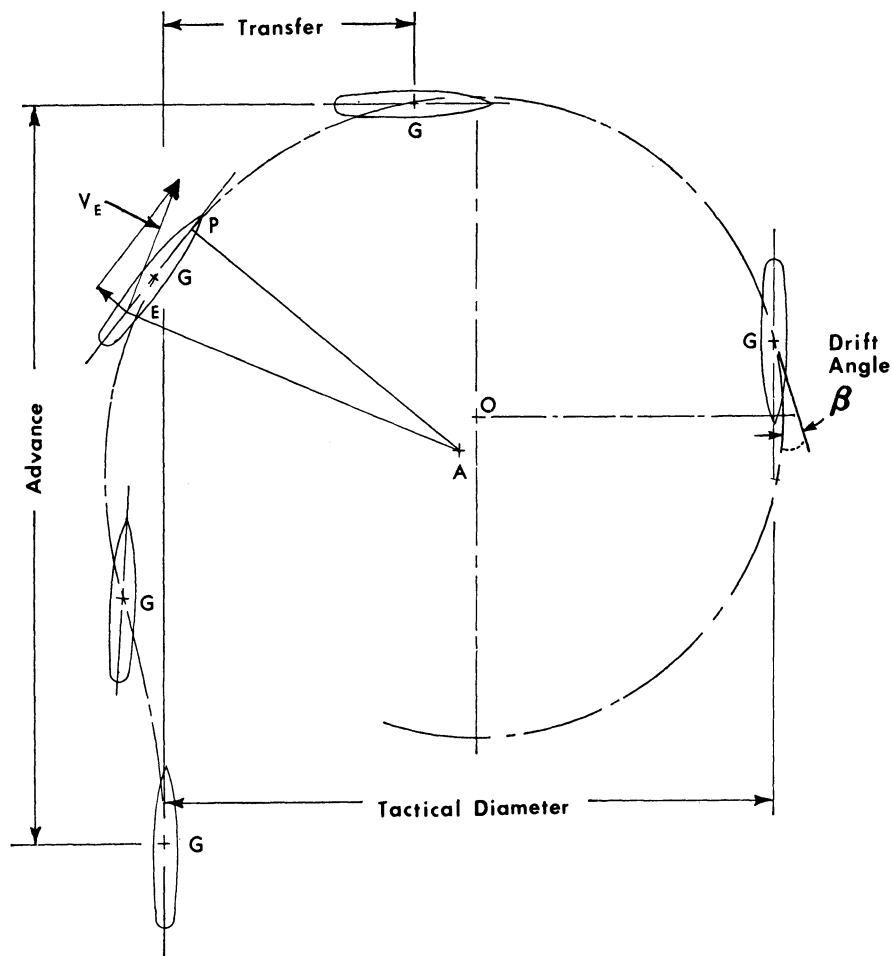


Figure 14-9. Motion of ship in turning

Most ships steer erratically in a following sea because the decreased relative speed between waves and ship allows any athwartship component of the wave force a greater time interval to act on the stern. Also, because of the orbital motion of water particles in a sea wave (see chapter 13), the rudder force will fluctuate between high and low values, and in one wave cycle it may even go from positive to negative values. This phenomenon is most apparent when the ship is moving in the same or nearly the same direction as the predominant wave system, where the speed of the wave system is only slightly greater.

Wind striking the ship from an angle will tend to blow the ship off course. Ships with high forecastles are very difficult to turn into the wind.

Steering is also influenced by the depth of water, narrow channels, number of propellers, and relative position of rudders astern of the propellers. Propellers extending below the keel line greatly increase the resistance to turning.

TURNING

When a ship is under way on a straight course in still water, the streamline flow about the rudder is generally

symmetrical with respect to the centerline plane, with no athwartship component of the water force. When the rudder is moved from its mid-position, the symmetry is disturbed, and a force acting at the center of pressure of the rudder is introduced. This results in a path of motion of the ship's center of gravity (figure 14-9) and in a heeling of the ship. The initial motion of translation is a spiral curve, which steadies to a circular path when the ship has turned approximately 90°.

The distances moved by the center of gravity, in the direction of the original course and at right angles to the original course, from the point where the rudder is started over until the heading has changed 90°, are named *advance* and *transfer*, respectively. The distance at right angles to the original course gained by the center of gravity when the ship turns 180° is the *tactical diameter*.

In turning, the ship moves with the bow inside and the stern outside the tangent to the path of the center of gravity. The angle between the tangent to the turning circle and the centerline of the ship is called the *drift angle* (β).

When the ship assumes its drift angle, there is a point on the centerline between the bow and the center of gravity at which the resultant velocity (rotation plus

translation) is directed along the centerline (that is, no component is at right angles to the centerline). To the observer aboard, the ship appears to rotate about this point, which is called the *pivot point*. The pivot point is normally located about one-third to one-sixth of the distance from the bow to the center of gravity. The *rudder angle* (δ_R) is the angle between the centerline of the ship and the rudder.

In figure 14-9, point *A* is the instantaneous center of the turning path when the ship is in the position indicated. The instantaneous velocity V_E of point *E* is normal to the radius *AE*. It should be evident that the instantaneous velocity of point *G*, being normal to the radius *AG* (not drawn), will be tangent to the turning path. Further, it should be evident that the instantaneous velocity of point *P* will lie on the centerline of the ship, inasmuch as the radius *AP* is normal to the centerline. The point *P* is the pivot point. When the ship has steadied in a circular path, the instantaneous center becomes fixed at point *O*.

14-5 Forces on a Ship

When the rudder is put over to an angle δ_R , a rudder force acting at the center of pressure of the rudder is generated. This force may be resolved into components normal to and along the rudder surface. The force parallel to the rudder surface has little effect on the turning action and may be disregarded. Considering figure 14-10, the rudder force *P* is shown acting normal to the rudder. This force may be resolved into an equal and parallel force P_2 and a couple *PD* acting at a point *C*. Point *C* is in the horizontal plane containing the rudder's center of pressure and is directly below the ship's center

of gravity. The force P_2 may be further resolved into $P_2 \cos \delta_R$ and $P_2 \sin \delta_R$, which are perpendicular and parallel to the centerline respectively.

1. *Initial phase.* The initial phase of the turn begins when the rudder is put over. Under the influence of force $P_2 \cos \delta_R$, the ship accelerates in an outward drift away from the center of turning (to port in figure 14-10). The force $P_2 \sin \delta_R$ opposes the motion, and a reduction of speed results. At the same time, under the influence of the couple *PD*, the ship begins to rotate about a vertical axis through *C*. This rotation is in the direction of the desired turn, as indicated in figure 14-10. The initial phase is over as soon as the rudder has overcome the ship's rotational inertia, and the ship has rotated through an angle that allows hydrodynamic forces to develop.

2. *Second phase.* As the ship rotates, the direction of the instantaneous velocity V_E rotates through an angle of attack β from the ship's centerline. The force P_2 and the couple *PD* continue to act, but the pressure distribution about the hull changes. The translation of the ship with the angle of attack β produces a pressure increase all along the outward side of the ship (away from turning center *O*), thus creating a force *R* acting at some point *B* and inclined at an angle α to the ship's centerline. This force may be resolved into an equal and parallel force R_1 and a couple *RM* acting at point *C*. The force R_1 may be further resolved into *lift* and *drag*, which are perpendicular and parallel to the direction of the instantaneous velocity. Note that lift and drag are defined in exactly the same way as they are on propeller blades and rudders because of the essential foil shape of the ship moving through a fluid medium at a specific angle of attack.

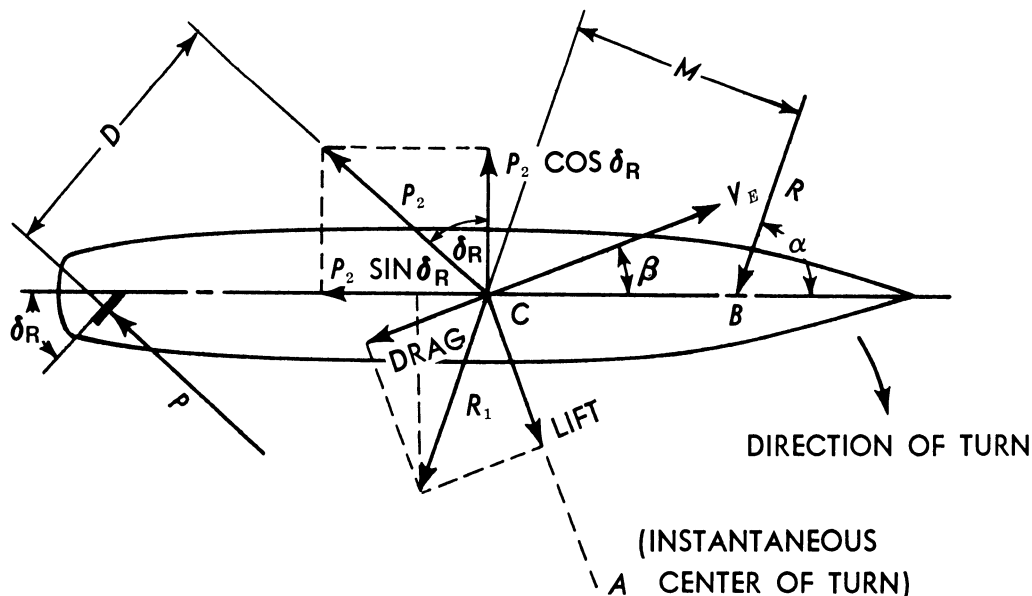


Figure 14-10. Forces on a ship caused by rudder action

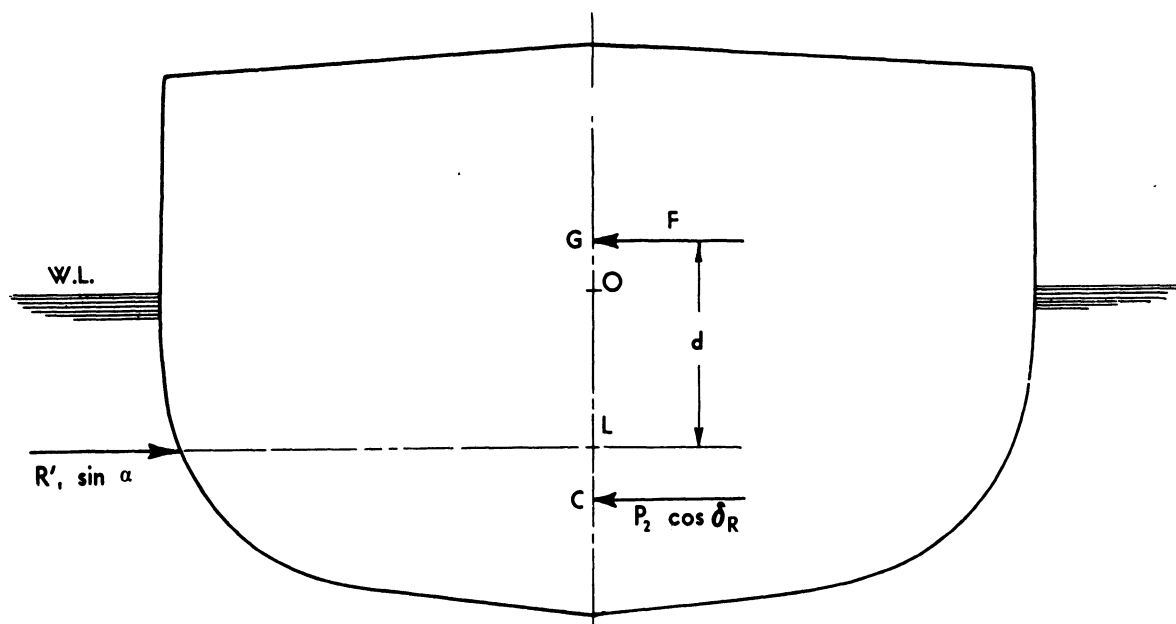


Figure 14-11. Heeling caused by rudder action (viewed from aft)

During the second phase, the couple RM aids the couple PD , and the turning motion is accelerated (if B is forward of C , which is the usual case). As this occurs, the ship's angle of attack increases, point B moves aft, and the turning tendency of the couple RM is reduced. This accounts for the S-shaped path during the first 90° of the turn. During this phase, the ship is also being influenced by the lift and drag components of R_1 . The ship is being further slowed by the drag and accelerated toward the instantaneous center of the turn. Thus the ship, which accelerated outwardly in the initial phase due to $P_2 \cos \delta_R$, begins to accelerate inwardly as the lift becomes greater than $P_2 \cos \delta_R$. It can be seen that the second phase is a transition between the initial phase and steady turning.

3. *Steady turning phase.* Eventually B moves abaft the center of gravity, and equilibrium is established between the rudder couple PD and the lift couple RM . The ship settles, then, on the circular portion of the turning path. The forces P_2 and R_1 continue to act at constant values. The ship continues to be slowed and accelerated toward the center of the turning circle. In the steady turning condition, this inward acceleration becomes constant, and the ship maintains a turning path of constant radius.

In discussing the heeling forces on a ship in a turn, it is necessary to consider a component of R_1 other than lift and drag. This is $R_1 \sin \alpha$, the component perpendicular to the ship's centerline. In figure 14-11, $R_1 \sin \alpha$ is assumed to act at L , the geometric center of the immersed centerplane, and produces an inward force that opposes the outward drift, due to $P_2 \cos \delta_R$. These forces, because of their lack of vertical agreement, re-

sult in a heeling action. The discussion of heeling action will follow the same three phases discussed above.

1. *Initial phase.* During the short time interval immediately after the rudder has been put over, the transverse component of rudder force $P_2 \cos \delta_R$, acting at the center of pressure of the rudder, produces an inward heeling moment whose arm may be assumed to be $\overline{OC}$. (This is a fair assumption for most ships, because at small angles of heel, the heeling axis is very close to the waterplane and near the center of gravity.) The moment of rudder force at this phase, acting as a heeling moment, is $\overline{OC} P_2 \cos \delta_R$.

2. *Second phase.* The component of the force on the hull $R_1 \sin \alpha$ gradually builds until it becomes greater than $P_2 \cos \delta_R$. As it builds, a heeling moment counter to $\overline{OC} P_2 \cos \delta_R$ develops, and the net heeling moment in the second phase of the turn is

$$M_1 = P_2 \cos \delta_R \overline{OC} - R_1 \sin \alpha \overline{OL}$$

This net moment results in an inward heel, because at the beginning of the turn, $R_1 \sin \alpha$ is zero. As $R_1 \sin \alpha$ begins to grow in magnitude, the inward heel diminishes and an outward heel may begin (usually in warships and higher-powered ships of ample freeboard).

3. *Steady turning phase.* The steady turning phase can best be analyzed by considering the centrifugal force resulting from the ship's inward acceleration toward the center of the turn. When the ship has settled into a steady (circular) turning path, the centrifugal force of her mass is acting at the ship's center of gravity G . This is the amount

$$\frac{\Delta v^2}{gr} \text{ (in tons)}$$

The ship, in a steady path, indicates that a state of equilibrium has been established between the outboard forces at G and C and the inward resisting force at L . According to figure 14-11, the centrifugal force would be acting counterclockwise about point L while the transverse component $P_2 \cos \delta_R$ is still acting in a clockwise direction. Thus, taking our moments about L , we have the net heeling moment

$$M = \frac{\Delta v^2}{gr} \overline{GL} - P_2 \cos \delta_R \overline{CL}$$

The greatest angle of heel is reached immediately after the change from inward to outward heel because, due to its mass inertia, the ship rolls past the position of equilibrium (figure 14-12). If, in a high-speed turn, the rudder is returned amidship at this instant, the inward heeling moment disappears and a dangerously large outward heel may result. This fact deserves some attention since the helmsman, fearing too large a heel, might be inclined to remove the rudder angle, whereas the only safe action would be to reduce speed.

At the equilibrium angle of heel, the algebraic sum of the three heeling moments is equal to the ship's righting moment. The angle of heel is dependent on the relative location of points C , L , and G of figure 14-11 and on the rudder angle, speed, and metacentric height.

14-6 Directional Control Systems at Low Speeds

All control surfaces, such as rudders and stabilizing fins, are by design forms of hydrodynamic foils. As such, they depend for their effective control force on the relative motion of the water over their surface. When a ship is maneuvering in close quarters in a harbor at low speed or is under way with no way on, the necessity for directional control still exists. As ships have grown larger and more unwieldy owing to their greatly increased inertial mass, the task of maneuvering them in restricted water at low speeds has required the services of small, relatively high-powered tugboats acting at various locations along the ship's length. This process is not as efficient nor as dependable as the ship's own self-contained systems, if she is able to supply these



Figure 14-12. The USS *Enterprise* (CVAN 65) making a high-speed turn. Note the heeling angle, which can become operationally dangerous when the forces involved are not understood.

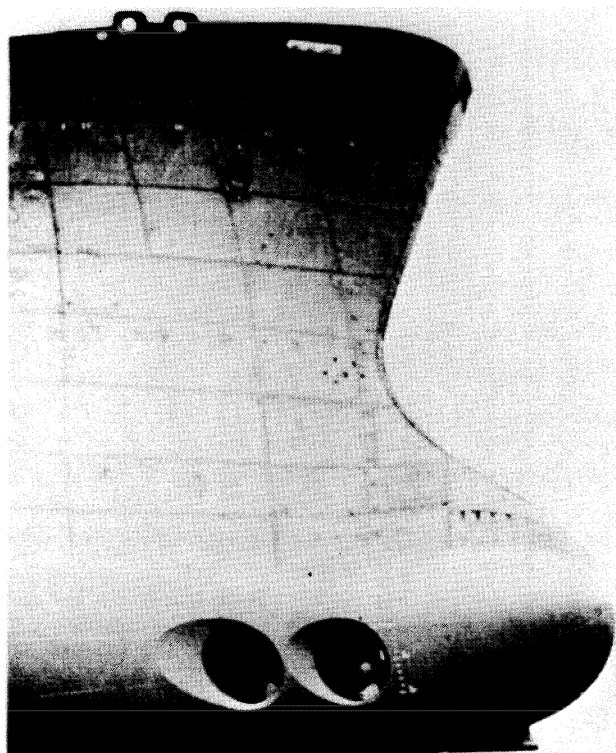


Figure 14-13. Twin L-thruster installation in a bulbous bow. Note that the aft entrance is in the “shadow” of the forward entrance. Longitudinal axes of the downstream fairings are oriented on the flow line as determined by a paint test. (From Norby and Ridley 1980.)

localized and directional thrusts. Consequently, some years ago directional control systems called *bow thrusters* were introduced. The original bow thrusters consisted basically of ducted or tunneled propellers, where the duct or tunnel was a transverse tunnel through the ship near the bow in a deep V-shaped, narrow, forward section. Within this bow tunnel a reversible propeller or propellers were installed, which, operating as propeller pumps, threw a relatively large mass flow of water to either side. This is essentially the nature of currently used bow thrusters, which are becoming increasingly popular and are designed into many ships, both large and small (see figure 14-13).

Bow-thruster technology was significantly improved with the development of controllable-pitch lateral thrusting units (*L-thrusters*) in the late 1950s. These units, using A.C. electric motors and located in an athwartship tunnel, are the most popular directional control device for applications demanding relatively high power. Fixed-pitch L-thrusters are still popular in the low power range, where D.C. electric motors or diesels can be used to vary the rpm of the thruster. L-thrusters located at both bow and stern have become almost standard equipment for vessels making frequent port calls, such as ferries and the Great Lakes bulk carriers of the 1960s. Oil tankers that load and discharge their

cargo at single buoy moorings require precise positioning without the help of tugs, and L-thrusters have also become standard equipment on such vessels. Roll-on, Roll-off (*Ro/Ro*) ships that make port calls in many harbors without adequate tugboat service have recently replaced ferries as the leader in the number of installations of bow and stern thrusters.

Rotating thrusters (*Ro-thrusters*) that consist of an open propeller or ducted propeller mounted on a pod, whose direction can be controlled through 360°, are becoming quite popular on offshore supply and support vessels, both regular and semisubmersible. Some Ro-thrusters are designed to be retractable into the hull in order to reduce drag at high speeds, but hull-mounted units can be lined up with the longitudinal axis of the ship to gain additional thrust during normal ship operations. Since most thruster applications require low pitch for bollard pull (zero forward velocity) situations, a controllable-pitch propeller on hull-mounted Ro-thrusters is essential for thrust augmentation at cruising speeds (Norby and Ridley 1980).

14-7 Measurements of Maneuvering Performance: Model Tests and Full-Scale Trials

The ITTC proposed a new maneuvering-trials code for full-scale trials in 1975. These maneuvering tests are performed when required during the builders' sea trials for the purpose of evaluating maneuverability characteristics of ships. The trial code is as follows:

General program

1. *Turning circles.* Turning circles should be performed to port and starboard up to 540° change of heading, at full speed with maximum rudder angle and with a rudder angle of 15°.

Additional turning circles at medium and low speeds should be considered.

2. *Pull-out maneuvers.* A pull-out maneuver, in which the rudder is simply returned to midship, should normally be performed when finishing the 15° turning trials, always to port as well as to starboard. It may alternately be made at the end of a zig-zag maneuver.

3. *Turning tests from zero speed.* Additional turning tests should be performed to both port and starboard from zero speed, using maximum rudder angle and engine half ahead. The test is complete when the heading has changed by 180°.

4. *Zig-zag maneuvers* (figure 14-14). A standard-type 10° helm/10° nominal-change-of-heading zig-zag maneuver should be performed at full speed of approach, preferably starting with a starboard helm execution.

When possible, one or two “modified” zig-zag maneuvers (that is, the 20°/10° and the 10°/10° maneuvers) should also be performed.



Figure 14-14. The USS *Long Beach* testing high-speed turns during trials

5. *Direct and reverse spiral tests.* In case the ship is likely to be dynamically unstable or only marginally stable, a spiral test should be performed to furnish the steady-state steering characteristics, if wind and sea-state conditions are favorable. The test procedure may follow “normal” or “reversed” routine, the latter being preferred for the unstable ship.

6. *Change-of-heading tests.* Special change-of-heading tests, useful for navigation purposes, may be performed for a range of approach speeds and rudder angles. Some of this information may be obtained from the initial transients of turning and zig-zag tests.

7. *Stopping tests.* Stopping test(s) with engine(s) reversed to full astern should be performed for one or more speeds of approach, rudders being kept amidship.

Whereas the crash-stop maneuvers from full ahead may often be required as an acceptance test, a stopping test at a low speed of approach is here recommended.

Special program extension for ships with lateral thrust units

1. *Turning maneuvers.* Turning maneuvers should be performed to port and starboard, completing 90° change of heading with full output of the thruster and main rudders amidship, all in a range of speeds between zero and 8 knots. The approach should be made with bow directly into the wind.

2. *Zig-zag maneuvers.* A zig-zag maneuver should be performed, switching the full output of the thruster

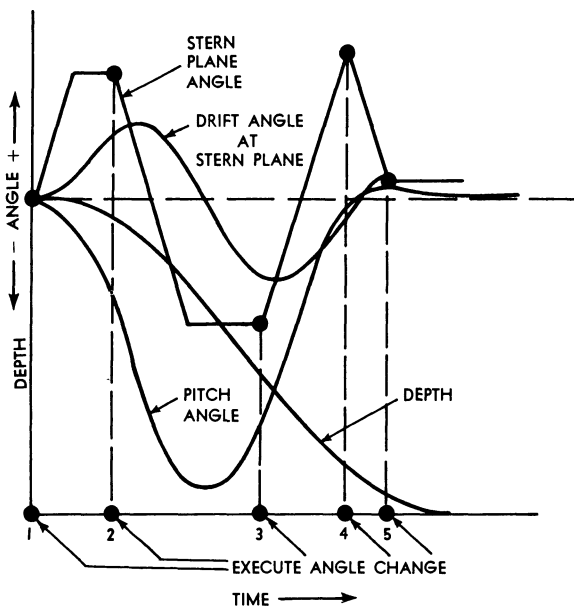


Figure 14-15. Submarine time-depth change history

at 10° change of heading, with the main rudders kept at midship, all at an approach speed in the range of 3 to 6 knots.

For further details, see appendix I of the Maneuverability Committee Report to the Fourteenth ITTC.

Model-ship correlation studies show reasonably good agreement between free-running (radio-controlled) model tests and full-scale experiments for slender ship forms. For full-bodied forms there are distinct differences between model and ship maneuvering experiments, which are generally attributed to scale effects on the separation of flow around the ship model during turns. So-called *captive model tests* are also performed in model tanks, in which forced oscillations are imposed on the model in the *horizontal planar motion (PMM) tests* in order to measure the maneuvering coefficients in the equations of motion. These coefficients can also be measured during a forced turn in a rotating arm basin.

14-8 Dynamic Control of Submarines

As mentioned in chapter 9, static depth and trim control must be used below about 3 knots, since the stern planes and fairwater or bow planes are not effective at low speeds and may even exhibit control reversals. Above 3 knots, the planes can generate sufficient normal forces to control the depth and trim of a "close in trim" submarine condition. Recall that the fairwater planes are lifting surfaces located reasonably close to the center of gravity of the submarine, so that their primary purpose is to control small changes in depth and their secondary purpose is trim control. The stern planes are located well aft of the center of gravity of the submarine

and their primary purpose is attitude (trim) control, similar to that of the elevators on an airplane.

The shape, area, and location of these control surfaces depend upon the size, speed, and other operational requirements of the submarine, and a detailed discussion of them must be left to a more specialized treatment of submarine design. However, the nature of the basic problems of vertical control may be cited briefly by examining figure 14-15, which shows the sequence and time history of a submarine in a simple change-of-depth maneuver. It can be seen in particular that the pitch angle, which is the submarine's attitude relative to the horizontal, progresses through a cyclic change from negative to positive angle throughout the process of changing depth. Thus it changes from down angle to up angle at about half of the depth change, so that while the submarine is still increasing depth during the last half of a dive, it is at an up angle. The pitch angle and depth finally level off together with the stern plane angle, which maintains, together with pitch angle, a slightly positive value at the increased depth (this is to counter the tendency of the boat to sink further because of compressive buoyancy change).

Directional control is provided by a pair of rudders above and below the stern, as shown in figure 14-16. The heeling moment in a turn would be balanced by the rudders were it not for the influence of the fairwater (sail), which extends vertically above the submarine hull. During a high-speed turn, the sail acts as an additional lifting surface, tending to roll the submarine inward. The top portion of the rudder is frequently larger than the bottom portion in order to partially balance out the influence of the fairwater (figure 14-16).

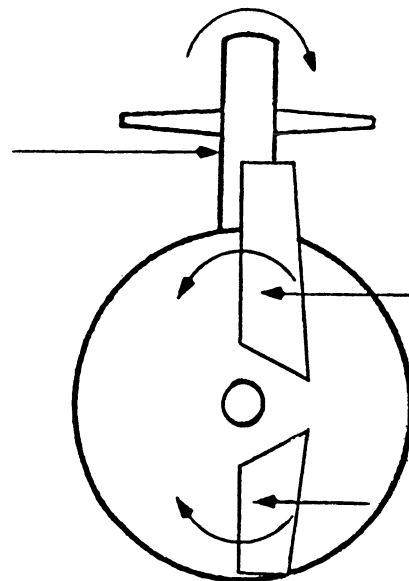


Figure 14-16. Forces causing roll during speed submarine turns

14-9 Control by Automation: Its Impact on Design

It became apparent to ship operators in the early part of this century that manual steering and course keeping were both costly and less effective than desired. Such steering required at least two helmsmen per watch. The human response mechanism is unable to produce anything other than an erratic track through the water when a ship is in a seaway. Because of these problems, there came into being the first of all ship automated systems, the *auto pilot*, or as the seaman of the day referred to it, the "iron mike."

This comparatively simple steering-control system consists, in its simplest form, of a course indicator together with a course-set device that signals any discrepancy in either direction to a relay. This starts the steering mechanism that responds with an appropriate rudder angle until the discrepancy is eliminated and the signal ceases (figure 14-17). In the years since the auto pilot's introduction, many sophistications have been added, such as feed-backs indicating the rate of the change of the heading. This allows the steering mechanism to anticipate the magnitude of the correction and to correct for overswing with the easing and, if necessary, reversing of the rudder angle.

This comparatively simple error-response system for automatic steering, however, is not the same as *automatic ship control*. Ship control, in the current systems of automation, is a far broader concept, which involves reducing the number of personnel required for manual operation. Ship control not only performs the routine tasks of course keeping, but also provides a system for remote control of propulsion and auxiliary machinery from the bridge or the control center. This includes

automated control of most traditionally manned stations for main engine operation, with control for boilers, feed water, fuel oil, and auxiliaries that are pertinent to speed and maneuvering changes through all ranges of ahead, stop, and backing procedures. It further includes monitoring of the systems and, on some vessels, the central controlling of docking and line-handling winches.

What such control involves can be better understood by the systems diagram and analysis shown in figure 14-18. This illustration is a basic study for elemental automatic control. The customary scope of naval architecture does not normally include the design of machinery or such internal systems, and these details and functions will, accordingly, not be treated here. However, the impact of automatic ship control on the design of ships must not be lightly passed over.

The primary motivating factor of ship automation is, of course, centered in the economics of operation and maintenance. The reduction of crew, both for naval vessels and commercial vessels, is of vital concern in the successful continuance of ship operation. Automation is the only rational solution to the labor problem for ships of conventional missions and configurations. The reduction of crew size is an assurance of continued economic vitality; this factor alone, which reflects the increase of overbearing operating costs of a naval fleet, must be a primary consideration for the continuance of a merchant fleet.

The designers of ships are, of course, acutely aware of the enormously increased costs of ships with complex automated systems. They must also be aware of the requirements imposed on them both by the command

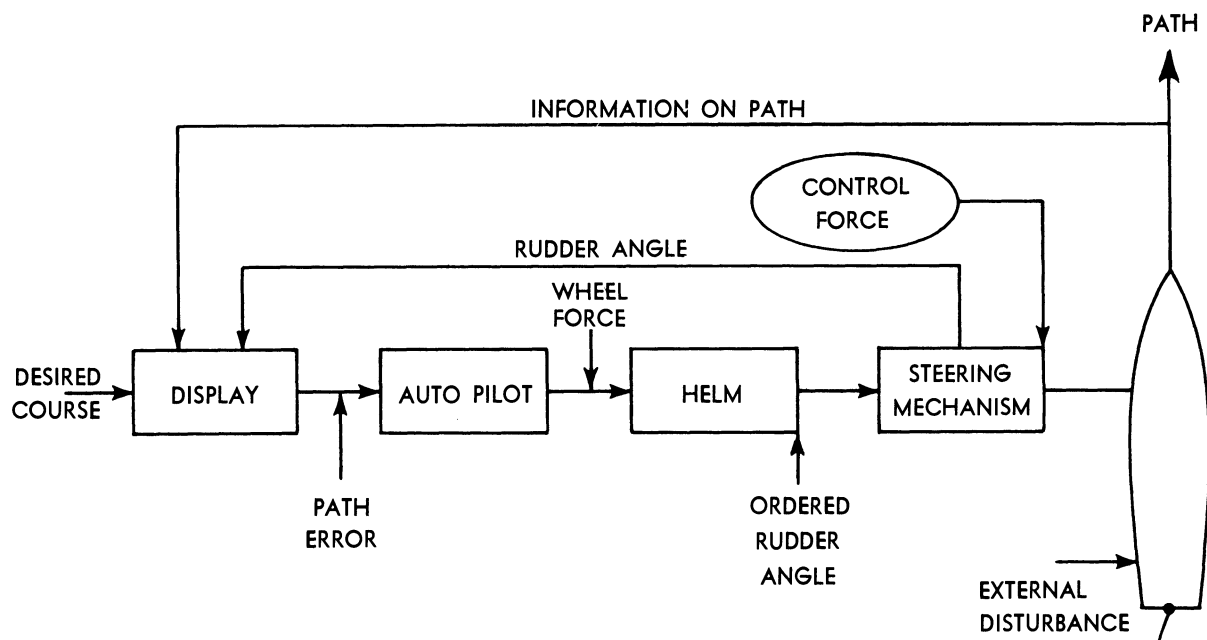


Figure 14-17. Simple ship control system (typical flow pattern)

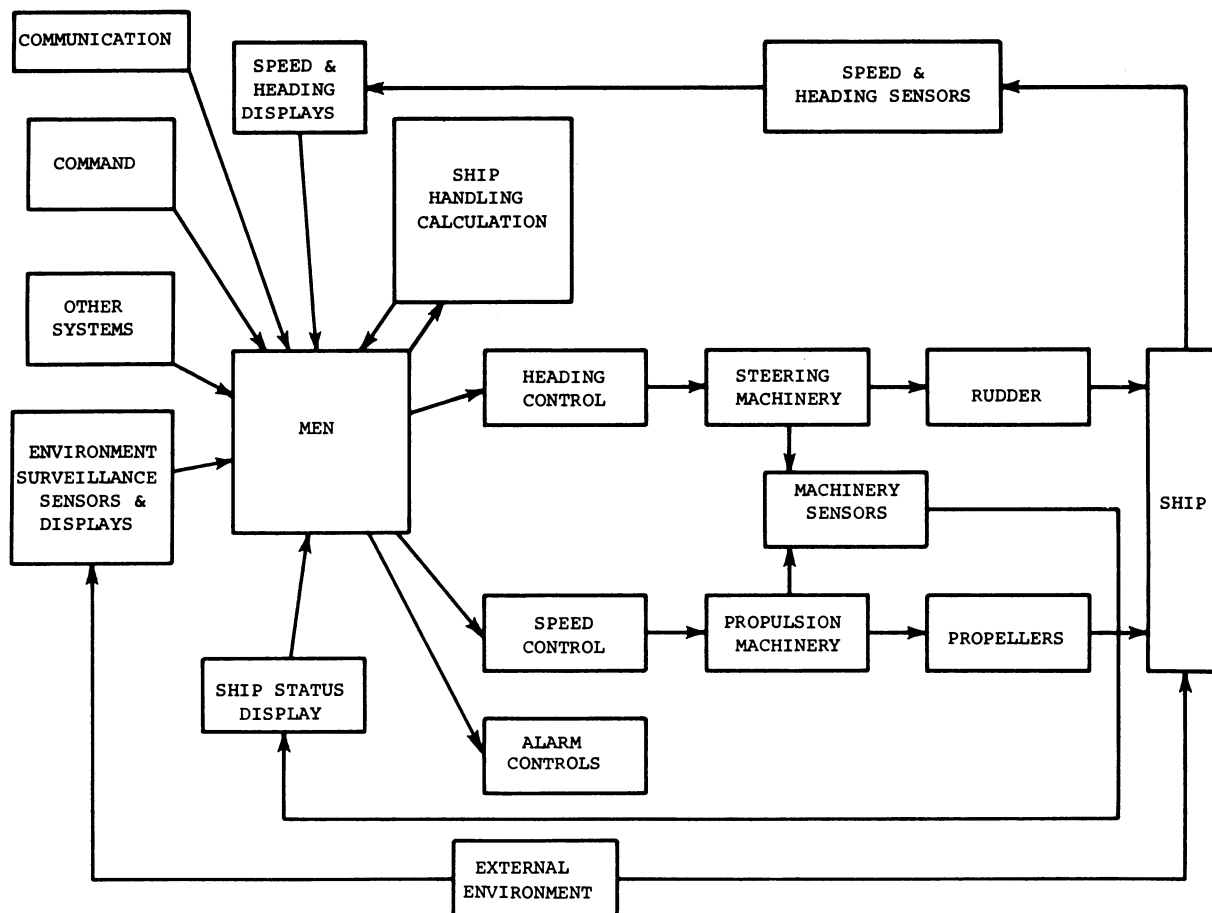


Figure 14-18. Typical destroyer-type ship control system

control of such ships and by the competitive rationalization of such costs. Such ships must be given every advantage for improved hydrodynamic character and responsive maneuverability through the *effective* design and application of the elements cited earlier in this chapter for directional control. They must have improved control stations derived from the increased utility of space aboard the ships. (This can, to a large extent, be accomplished through the reduction in living space required by previously large crews.) Thorough engineering designs prior to construction must include the most efficient accommodation of these control systems, particularly by taking advantage of the new shipbuilding "block" methods, where all of the interior systems are complete and included in the subassembly process.

Finally, the designer must increase the tempo of his

traditional search for more economical construction, whether it be in new materials, new design techniques, structural simplifications, or improved configurations.

The major components of steam-turbine propulsion machinery, including the fuel oil system, the boiler with its feed system and complex piping, and high- and low-pressure turbines with astern elements, obviously require a sophisticated system for automatic control. Primarily because of this, together with required warm-up times and lead times for increased boiler output, the steam plant is not as well adapted to automation as the diesel- or gas-turbine plant, where the energy transfer from fuel to engine is direct and power changes can be absorbed almost at once. Therefore automation is far more economical and effective when applied to diesel- and gas-turbine-driven ships.

APPENDIX A

Table of Fresh- and Saltwater Properties

Values of Mass Density ρ for Fresh and Salt Water

Values adopted by the ITTC meeting in London, 1963.
Salinity of salt water 3.5 percent.

Density of fresh water ρ , lb-sec ² /ft ⁴ (= slugs/ft ³)	Temp, deg F	Density of salt water ρ_s , lb-sec ² /ft ⁴	Density of fresh water ρ , lb-sec ² /ft ⁴	Temp, deg F	Density of salt water ρ_s , lb-sec ² /ft ⁴
1.9399	32	1.9947	1.9384	59	1.9905
1.9399	33	1.9946	1.9383	60	1.9903
1.9400	34	1.9946	1.9381	61	1.9901
1.9400	35	1.9945	1.9379	62	1.9898
1.9401	36	1.9944	1.9377	63	1.9895
1.9401	37	1.9943	1.9375	64	1.9893
1.9401	38	1.9942	1.9373	65	1.9890
1.9401	39	1.9941	1.9371	66	1.9888
1.9401	40	1.9940	1.9369	67	1.9885
1.9401	41	1.9939	1.9367	68	1.9882
1.9401	42	1.9937	1.9365	69	1.9879
1.9401	43	1.9936	1.9362	70	1.9876
1.9400	44	1.9934	1.9360	71	1.9873
1.9400	45	1.9933	1.9358	72	1.9870
1.9399	46	1.9931	1.9355	73	1.9867
1.9398	47	1.9930	1.9352	74	1.9864
1.9398	48	1.9928	1.9350	75	1.9861
1.9397	49	1.9926	1.9347	76	1.9858
1.9396	50	1.9924	1.9344	77	1.9854
1.9395	51	1.9923	1.9342	78	1.9851
1.9394	52	1.9921	1.9339	79	1.9848
1.9393	53	1.9919	1.9336	80	1.9844
1.9392	54	1.9917	1.9333	81	1.9841
1.9390	55	1.9914	1.9330	82	1.9837
1.9389	56	1.9912	1.9327	83	1.9834
1.9387	57	1.9910	1.9324	84	1.9830
1.9386	58	1.9908	1.9321	85	1.9827
			1.9317	86	1.9823

NOTE: For other salinities, interpolate linearly.

Values of Kinematic Viscosity ν for Fresh and Salt Water

Values adopted by the ITTC meeting in London, 1963.
Salinity of salt water 3.5 percent.

Kinematic viscosity of fresh water ν , $\frac{\text{ft}^2}{\text{sec}} \times 10^5$	Temp, deg F	Kinematic viscosity of salt water ν_s , $\frac{\text{ft}^2}{\text{sec}} \times 10^5$	Kinematic viscosity of fresh water ν , $\frac{\text{ft}^2}{\text{sec}} \times 10^5$	Temp, deg F	Kinematic viscosity of salt water ν_s , $\frac{\text{ft}^2}{\text{sec}} \times 10^5$
1.9231	32	1.9681	1.2260	59	1.2791
1.8871	33	1.9323	1.2083	60	1.2615
1.8520	34	1.8974	1.1910	61	1.2443
1.8180	35	1.8637	1.1741	62	1.2275
1.7849	36	1.8309	1.1576	63	1.2111
1.7527	37	1.7991	1.1415	64	1.1951
1.7215	38	1.7682	1.1257	65	1.1794
1.6911	39	1.7382	1.1103	66	1.1640
1.6616	40	1.7091	1.0952	67	1.1489
1.6329	41	1.6807	1.0804	68	1.1342
1.6049	42	1.6532	1.0660	69	1.1198
1.5777	43	1.6263	1.0519	70	1.1057
1.5512	44	1.6002	1.0381	71	1.0918
1.5254	45	1.5748	1.0245	72	1.0783
1.5003	46	1.5501	1.0113	73	1.0650
1.4759	47	1.5259	0.9984	74	1.0520
1.4520	48	1.5024	0.9857	75	1.0392
1.4288	49	1.4796	0.9733	76	1.0267
1.4062	50	1.4572	0.9611	77	1.0145
1.3841	51	1.4354	0.9492	78	1.0025
1.3626	52	1.4142	0.9375	79	1.9907
1.3416	53	1.3935	0.9261	80	0.9791
1.3212	54	1.3732	0.9149	81	0.9678
1.3012	55	1.3535	0.9039	82	0.9567
1.2817	56	1.3343	0.8931	83	0.9457
1.2627	57	1.3154	0.8826	84	0.9350
1.2441	58	1.2970	0.8722	85	0.9245
			0.8621	86	0.9142

NOTE: For other salinities, interpolate linearly.

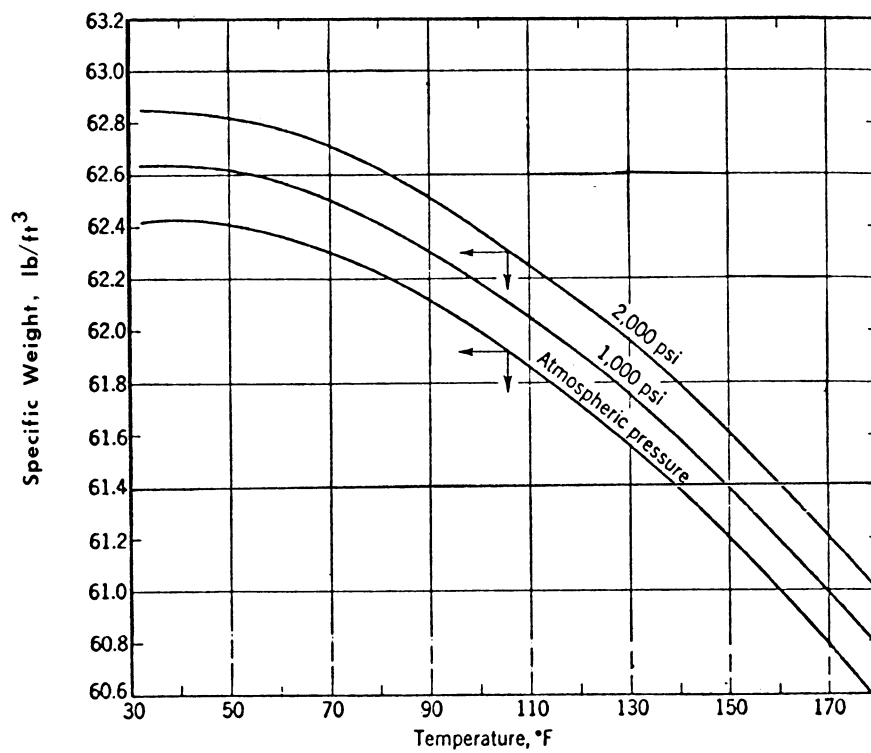


Figure A-1. Specific weight of fresh water

APPENDIX B

Numerical Integration Techniques in Ship Design

Computation of an Area by the Trapezoidal Rule

An integration rule frequently used in ship calculations is known as the *trapezoidal rule*. To find the area under a curve such as $ABCD$ in figure B-1, divide AB into any number of equal parts and at the points of division erect perpendiculars.

The area of a trapezoid is equal to one-half of the sum of the bases multiplied by the altitude. The area of the trapezoid $ADGH$ is therefore

$$A = h \left(\frac{1}{2}y_0 + \frac{1}{2}y_1 \right)$$

where h = the distance between ordinates.

If the ordinates are sufficiently close together, the curve will not vary appreciably from the chord, and the small area between D and G bounded by the straight line and the curve will be negligible. It is upon this assumption that the trapezoidal rule is based. The area of the second trapezoid is $h \left(\frac{1}{2}y_1 + \frac{1}{2}y_2 \right)$, the third is $h \left(\frac{1}{2}y_2 + \frac{1}{2}y_3 \right)$, and so on. It thus appears that in computing an area, every ordinate appears twice except the *end ordinates*. Summing up,

$$A = h \left(\frac{1}{2}y_0 + y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7 + \frac{1}{2}y_8 \right)$$

or in general,

$$A = h \left(\frac{1}{2}y_0 + y_1 + y_2 + \dots + y_{n-1} + \frac{1}{2}y_n \right)$$

where n is any integer. This expression is the trapezoidal rule.

To illustrate the use of this rule in the calculation of

an area, assume the ordinates of area $ABCD$ of figure B-1 are listed in table B-1 with an ordinate spacing of 10 feet. The solution of the area in table B-1 is self-explanatory.

Location of the Centroid of an Area

To find the location of the geometric center of an area (or its center of gravity), first find its moment about some reference axis. This overall moment must then be divided by the total area in order to find the arm or perpendicular distance of the center of gravity from the reference axis.

It must be pointed out before explaining an example that the use of the trapezoidal rule is less accurate in solving for centers of geometric figures than in solving for areas and volumes. Note that this rule was derived in the preceding section on the basis of trapezoidal areas. However, it may be applied to the solution of centers

Table B-1.

Station	Ordinate (feet)	Multiplier	Function of Area $f(A)$
Y_0	12	1/2	6
Y_1	14	1	14
Y_2	16	1	16
Y_3	16.5	1	16.5
Y_4	17	1	17
Y_5	16	1	16
Y_6	15	1	15
Y_7	13	1	13
Y_8	10	1/2	5

$$\Sigma f(A) = 118.5$$

$$h = 10 \text{ feet}$$

$$A = h \times \Sigma f(A) = 10 \times 118.5 = 1,185 \text{ square feet}$$

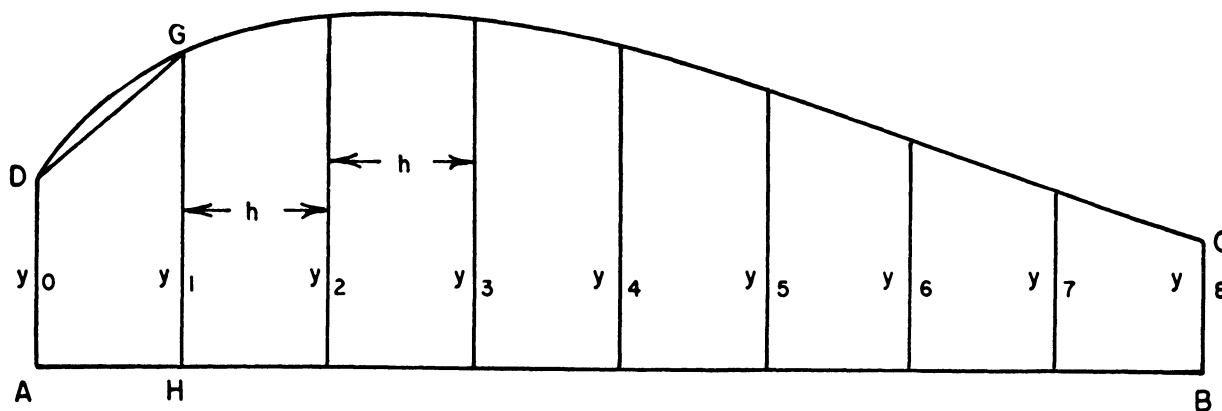


Figure B-1. Ordinates for an area

of areas as an approximate means of obtaining a mean moment from moments of incremental areas.

To illustrate, consider figure B-2 and the ordinate values listed in table B-2. For the reference axis, assume

Table B-2.

Station	Ordinate (feet)	Multiplier	Function of Area $f(A)$	Moment Arm about y_4	Function of Moment $f(M)$
0	0	1/2	0	+4	0
1	5	1	5.0	+3	15
2	10	1	10.0	+2	20
3	14	1	14.0	+1	14
					+49
4	16.5	1	16.5	0	0
5	17.8	1	17.8	-1	17.8
6	17.1	1	17.1	-2	34.2
7	15.3	1	15.3	-3	45.9
8	11.4	1/2	5.7	-4	22.8
			$\Sigma f(A) 101.4$		-120.7
					+49
				$\Sigma f(M) -71.7$	

$$\begin{aligned}
 h &= 10 \\
 A &= h \Sigma f(A) = 10 \times 101.4 \\
 &= 1,014 \\
 M_{y_4} &= h^2 \Sigma f(M) = 100 \times (-71.7) = 7,170 \\
 CG_{y_4} &= \frac{7,170}{1,014} = 7.05 \text{ feet to the right of } y_4
 \end{aligned}$$

in this case one of the midordinates, say y_4 , which obviously will be fairly near the centroid. (This selection is purely arbitrary.) Also establish, arbitrarily, that values to the left of this ordinate will be positive and to the right, negative. In this case, use the ordinate as a function of the area, and for each ordinate multiply it by the moment arm (distance of the ordinate from the reference axis). In each case the moment arm has a common factor: the spacing between ordinates. This factor can conveniently be omitted until the end and then be multiplied by the summation of the function of moments. Remember also, that since linear ordinates were used as functions of area, one must multiply the summation of the functions of moments again by the ordinate spacing to account for the fact that the technique involves summing up the moments of small sectional areas. This will make the summation of the moment about the line y_4 take the form of

$$M_{AD} = h^2 \Sigma f(M)$$

The distance of the CG from y_4 is the net moment of the area divided by the area.

$$CG_{AD} = \frac{h^2 \Sigma f(M)}{h \Sigma f(A)} = \frac{M_{y_4}}{A}$$

Table B-2 gives the solution for the distance of the center of gravity from line y_4 .

The centroid of waterplane areas, i.e., centers of

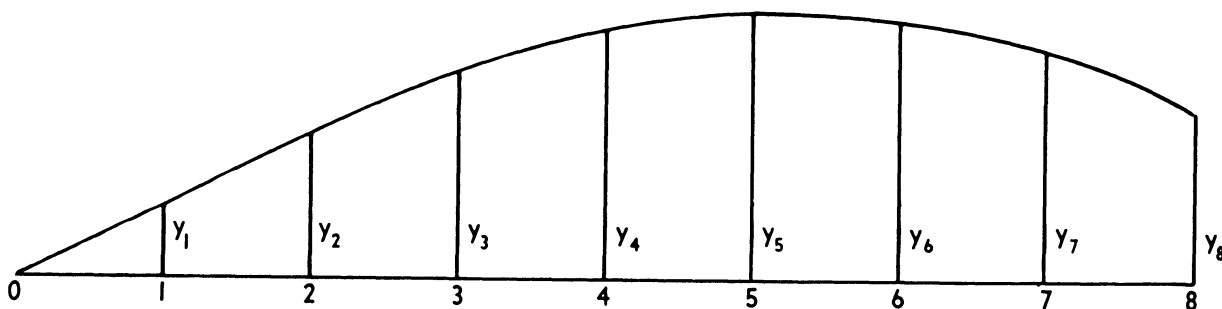


Figure B-2. Ordinates for an area and its CG

flotation, are computed according to the example discussed above. Since waterplanes are symmetrical areas, centers of flotation are located on the centerline axis, and only one reference axis need be used in defining them.

For unsymmetrical areas, a second reference axis must be used to define the point, just as two systems of coordinates are needed for the location of any point in a plane area.

Computation of a Volume

The calculation of an area as shown in the foregoing section is the first step in the computation for a volume. Because ships are irregularly shaped volumes, it is most convenient to pass reference planes, generally the waterlines or sections, and compute their areas. Those areas, which are at equally spaced intervals, may now be used as *ordinates* in computing the volume of the space through which they are passed. Suppose, for example, a tank, such as the one shown in figure B-3, in a bottom compartment or double bottom of a ship is bounded on one side and the bottom by the shell plating of the hull, on the ends by transverse bulkheads, on the top by a flat deck, and on the inboard side by a longitudinal bulkhead. Transverse sections are passed equal distances apart, giving areas such as *BCF* and *ADE* of figure B-3. The areas of each of these sections may be computed in the same manner as that of *ABCD* in figure B-1. They will now be used as ordinates for the computation of the volume.

To illustrate this, assume the transverse areas of figure B-2 are as listed in table B-3 with the sections spaced 10 feet apart. The computation is as indicated in table B-3.

To express this volume in tons of water (if the tank were filled), the volume would be divided by the appropriate density factor.

Computation of the Displacement of a Ship

The preceding sections indicate the two basic steps in calculating the displacement of a ship. In actual practice, however, the areas of each half-section up to the

Table B-3

Station	Ordinate (sq ft)	Multiplier	Function of Volume <i>f(V)</i>
0	22	1/2	11
1	38	1	38
2	60	1	60
3	68	1	68
4	72	1	72
5	62	1	62
6	50	1	50
7	32	1	32
8	16	1/2	8

$$\Sigma f(V) = 401$$

$$V = h \times \Sigma f(V) = 10 \times 401 = 4,010 \text{ cubic feet}$$

designer's waterline are obtained mechanically by use of a planimeter. The naval architect uses these values as ordinates and calculates the volume as in table B-3 or by some equally useful approximate integration method. The displacement of the ship at the designed draft in salt water is obtained by dividing the volume in cubic feet by $35 \frac{\text{ft}^3}{\text{ton sw}}$. In fresh water, the volume

of displacement would be divided by $36 \frac{\text{ft}^3}{\text{ton fw}}$.

The displacement may also be calculated by obtaining the areas of equally spaced waterlines and using these as ordinates between the keel and the designed waterline. This method is not as accurate, however, as the use of section areas for ordinates. On the equivalent spacing, waterlines change shape much more rapidly than do sections, making a greater error in the approximate integration. Also, waterlines are too extensive in area to be convenient for mechanical measurement of the area by a planimeter.

Displacements are calculated over a range of drafts through which the ship is expected to operate. From these, curves of displacements against drafts are drawn for both fresh and salt water.

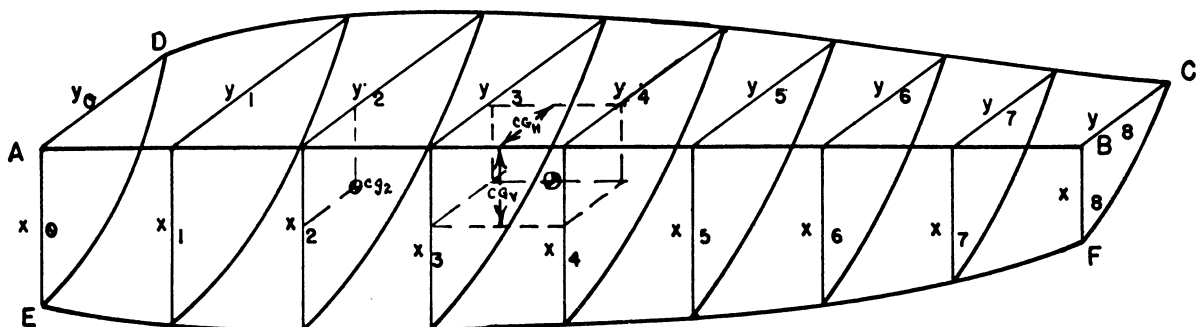


Figure B-3. Ordinates for center of gravity of a volume

Location of the Center of a Volume

Locating the center of gravity of a volume of uniform density simply involves further development of the foregoing discussion. Since the work will now be done in three dimensions instead of two, the distances of the center of gravity from three reference axes or planes will be required in order to completely define the point.

First, one must pass equally spaced transverse sections through the volume giving plane areas such as ADE and BCF of figure B-3. In order to obtain the moment arms of each plane surface, the location of the centroids of each area must be computed as outlined above. Using the areas of these planes as ordinates, the volume is computed. The moment of the volume about each of the three reference axes divided by the volume will give the distance of the center of gravity from that axis. The process briefly outlined here is a long and tedious exercise.

To illustrate as simply as possible, assume that the areas of the transverse sections are known and the location of the centroid of each has been determined by the method outlined just previously. This will reduce the numerical work in this example to the minimum necessary to illustrate the method of locating the center of gravity of a volume. Also, work from some centrally located ordinate for one reference axis as indicated above. The volume for this illustration will have the values listed in table B-3 for the computation of a volume. The reference axes or planes are arbitrarily selected as

1. The transverse vertical plane through y_4 , parallel to $\overline{ADE}$
2. The horizontal plane $\overline{ABCD}$
3. The longitudinal vertical plane $\overline{ABFE}$.

Assume that the center of gravity of each area has been located as listed in table B-4.

These distances from the reference axis to the centers of gravity will now be the moment arms which, when multiplied by the functions of volume, will become the functions of moment from their respective reference planes.

To summarize the procedure for the problem: first,

Table B-4.

Station	Centroid from ABCD (feet)	Centroid from ABFE (feet)
0	7.2	4.2
1	7.0	5.6
2	6.8	7.7
3	6.3	7.9
4	6.2	8.2
5	6.5	7.6
6	6.6	6.8
7	6.8	5.7
8	7.1	4.1

locate the center of gravity longitudinally in reference to some plane, in this case the plane containing x_4 and y_4 ; secondly, locate G in reference to a horizontal plane, in this case, plane $ABDC$; and finally, locate G in reference to a vertical plane, in this case, plane $ABFE$.

Before tabulating in the form, consider with reference to one ordinate what the moments will be. If a vertical plane through y_4 is the central reference plane, then, taking station 2 as the example,

$$\text{moment about } y_4 = y_2 \times 2h, \text{ where } y_2 \text{ is} \\ \text{the area of station 2}$$

$$\text{moment about } ABCD = y_2 \times 6.8$$

$$\text{moment about } ABFE = y_2 \times 7.7$$

(The values 6.8 and 7.7 are obtained from table B-4.)

The complete tabulation and solution is given in table B-5.

$$\begin{aligned} M_{y_4} &= h^2 \times \sum f(M)_{y_4} = 100 \times 56 = 5,600 \\ M_{ABCD} &= h \times \sum f(M)_{ABCD} = 10 \times 2,635 = 26,350 \\ M_{ABFE} &= h \times \sum f(M)_{ABFE} = 10 \times 2,874 = 28,740 \\ \therefore CG_{y_4} &= \frac{M_{y_4}}{V} = \frac{5,600}{4,010} = 1.40 \text{ feet to left of } y_4 \\ CG_{ABCD} &= \frac{M_{ABCD}}{V} = \frac{26,350}{4,010} \\ &= 6.57 \text{ feet below horizontal} \\ &\quad \text{reference plane} \\ CG_{ABFE} &= \frac{M_{ABFE}}{V} = \frac{28,740}{4,010} \\ &= 7.17 \text{ feet outboard of } ABFE \end{aligned}$$

The center of gravity of a volume is used in many ship computations, most frequently in determining the location of the *center of buoyancy*. The *center of buoyancy is the center of gravity of the immersed volume of the ship* and is located by giving its vertical distance from the keel and its horizontal distance from the midship section. Because the ship's hull is always symmetrical, the center of buoyancy must lie on the longitudinal vertical centerplane, and a third reference axis is unnecessary. Curves of vertical and longitudinal positions of centers of buoyancy are plotted against drafts in feet.

The method outlined in table B-5 is frequently used to locate the centers of irregularly shaped tanks. It should be realized that the term *center of gravity* of a volume as used in this discussion refers actually to the geometric center and is in no way related to the actual center of gravity of the ship or any system of weights.

Other Rules for Approximate Integration

The trapezoidal rule, as illustrated in the foregoing sections of this appendix, is used frequently for hand calculations, perhaps because it requires no specific

Table B-5

Station	Ordinate (sqft)	Multiplier	Function of Volume, $f(V)$	Arm about y_4	$f(M)$, Function of Moment about y_4	Arm about $ABCD$	$f(M)$, Function of Moment about $ABCD$	Arm about $ABFE$	$f(M)$, Function of Moment about $ABFE$
0	22	1/2	11	+4	+ 44	7.2	79	4.2	46
1	38	1	38	+3	+114	7.0	266	5.6	213
2	60	1	60	+2	+120	6.8	408	7.7	462
3	68	1	68	+1	+ 68	6.3	428	7.9	537
					+346				
4	72	1	72	0	0	6.2	446	8.2	590
5	62	1	62	-1	- 62	6.5	403	7.6	471
6	50	1	50	-2	-100	6.6	330	6.8	340
7	32	1	32	-3	- 96	6.8	218	5.7	182
8	16	1/2	8	-4	- 32	7.1	57	4.1	33
$\Sigma f(V) =$			401		-290	$\Sigma f(M)_{ABCD} = 2,635$		$\Sigma f(M)_{ABFE} = 2,874$	
					+346				
					$\Sigma f(M)_{y_4} = +56$				

number of sections and is adaptable to the gradual curvature in most ship waterlines. There are other rules having their own particular limitations and advantages that are also used in the same basic manner as the illustrations with the trapezoidal rule.

Simpson's first rule, commonly called Simpson's rule, requires, in contrast to the trapezoidal rule, that the figure or volume be divided into an even number (n) of parts. The area (or volume) is then considered to contain $n/2$ elements, each with a length of base equal to twice the common interval h between ordinates.

The basic assumption of Simpson's rule is that the curved side of the figure in an elementary segment (such as ABC of figure B-4) conforms to the equation of a second-order parabola. Thus

$$Y = ax^2 + bx + c$$

The areas under the curved portion may therefore be expressed mathematically, and from figure B-4

$$\text{area } \overline{OABCM} = \frac{1}{3}h(OA + 4BN + MC)$$

$$\text{area } \overline{MCDEK} = \frac{1}{3}h(MC + 4LD + KE)$$

$$\text{area } \overline{KEFGH} = \frac{1}{3}h(KE + 4JF + HG)$$

By addition, the total area of the figure is

$$\begin{aligned} \text{area } \overline{OAGH} \text{ (total)} = A &= \frac{1}{3}h(OA + 4BN \\ &+ 2MC + 4LD + 2KE + 4JF + HG) \end{aligned}$$

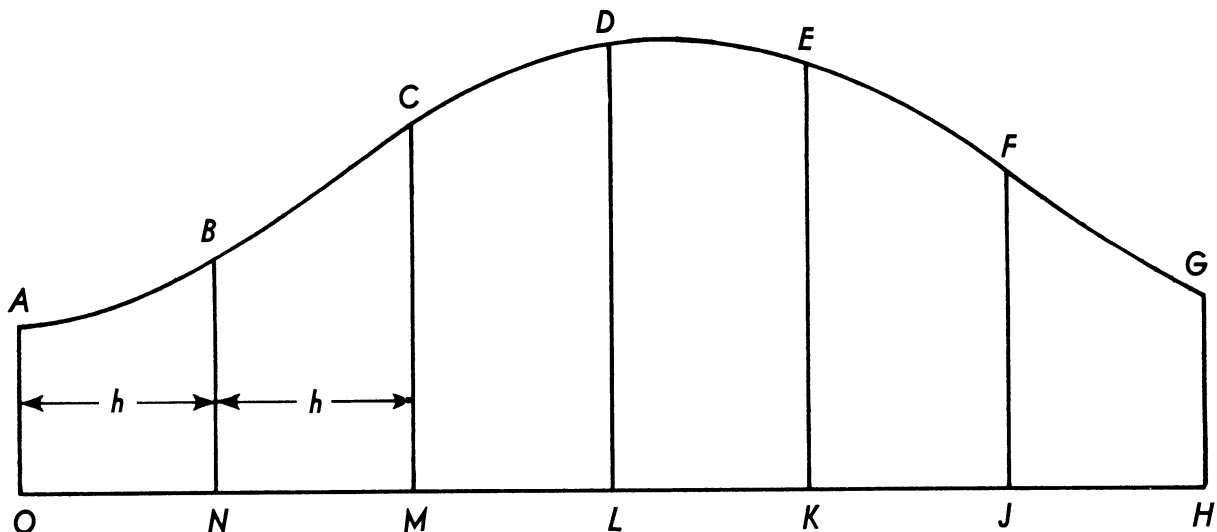


Figure B-4. Ordinates for Simpson's rule

The Simpson's rule multipliers are therefore of the sequence 1, 4, 2, 4, 2, 4, 1. The alternate multipliers—4 and 2—are used sequentially as many times as there are intermediate ordinates with unity as the multiplier for end ordinates.

As mentioned, this rule is accurate for areas bounded by parabolic curves and is approximate to the extent that the basic curvature approximates the parabola.

Simpson's second rule is developed from a third-order equation and can be reduced to the expression for an area (referring to the area in figure B-4) as

$$A = \frac{3}{8}h(OA + 3BN + 3MC + 2LD + 3KE + 3JF + HG)$$

In the case of this rule, the basic area or volume must be divided into a number of parts that make up a multiple of 3. The multipliers are apparent from the preceding equation.

There are other rules for approximate integration, such as the *five-eight rule* and *Tchebycheff's rule*, that are useful in specific cases but are not as generally applicable as those cited. In any case, all these rules for approximate integration are useful basic tools and are adaptable to computer programming and applications.

APPENDIX C

Conversion Table and SI Unit Prefixes

Conversion Table

<i>Multiply</i>	<i>by</i>	<i>to obtain</i>	<i>Multiply</i>	<i>by</i>	<i>to obtain</i>
atmospheres	33.90	feet of water	grams	10^3	milligrams
atmospheres	10.333	kgs. per sq. meter	grams	0.03527	ounces
atmospheres	14.70	pounds per sq. inch	grams	0.03215	ounces (troy)
centimeters	0.3397	inches	grams	0.07093	poundals
centimeters	0.01	meters	grams	2.205×10^{-3}	pounds
centimeters	10	millimeters	horse-power	44.44	B. t. units per min.
centimeters per second	0.6	meters per minute	horse-power	33.000	ft-lbs per min.
centimeters per second	0.02237	miles per hour	horse-power	550	ft-lbs per sec.
cubic centimeters	3.531×10^{-5}	cubic feet	horse-power	745.7	watt(w)
cubic centimeters	6.102×10^{-3}	cubic inches	horse-power	1.014	hp (metric)
cubic centimeters	10^{-4}	cubic meters	horse-power metric	735.50	watt(W)
cubic centimeters	2.642×10^{-4}	gallons	inches	2.540	centimeters
cubic centimeters	10^{-3}	liters	inches of water	25.40	kgs per sq meter
cubic feet	62.43	pounds of water	inches of water	5.204	pounds per sq ft
cubic feet	2.832×10^4	cubic cms	kilograms	10^3	grams
cubic feet	1728	cubic inches	kilograms	2.2046	pounds
cubic feet	0.02832	cubic meters	kilogram force (kgf)	9.807	Newton (N)
cubic feet	7.481	gallons	kgf per square meter	9.807	pascals (Pa)
cubic feet	28.32	liters	kgf per square centimeter	9.8067×10^4	pascals (Pa)
cubic feet per minute	472.0	cubic cms per sec	kilometers	10^5	centimeters
cubic feet per minute	0.1247	gallons per sec	kilometers	3281	feet
cubic feet per minute	62.4	lbs. of water per min.	kilometers	10^3	meters
cubic inches	16.39	cubic centimeters	kilometers	0.6214	miles
cubic inches	5.787×10^{-4}	cubic feet	knot	1.688	ft per sec
cubic inches	1.639×10^{-3}	cubic meters	$\log_{10} N$	2.303	$\log_e N$ or $\ln N$
cubic inches	4.329×10^{-3}	gallons	$\log_e N$ or $\ln N$	0.4343	$\log_{10} N$
cubic inches	1.639×10^{-3}	liters	meters	100	centimeters
degrees (angle)	60	minutes	meters	3.2808	feet
degrees (angle)	0.01745	radians	meters	39.37	inches
feet	0.3048	meters	meters	10^{-2}	kilometers
feet of water	0.02950	atmospheres	meters	10^2	millimeters
feet of water	304.8	kgs per sq meter	miles	1.609×10^5	centimeters
feet of water	62.43	pounds per sq ft	miles	5280	feet
foot-pounds	5.050×10^{-7}	horse-power-hours	miles	1.6093	kilometers
foot-pounds	0.1383	kilogram-meters	miles	1760	yards
foot-pounds per minute	3.030×10^{-5}	horse-power	<i>Multiply</i>	<i>by</i>	<i>to obtain</i>
foot per second	0.592	knots	miles (nautical)	1852	meters (m)
<i>Multiply</i>	<i>by</i>	<i>to obtain</i>	miles per hour	44.70	centimeters per sec
gallons	0.1337	cubic-feet	miles per hour	88	feet per minute
gallons	3.785×10^{-3}	cubic meters	miles per hour	1.467	feet per second
gallons	3.785	liters	miles per hour	1.6093	kilometers per hour
grams	10^3	kilograms	miles per hour	0.8684	knots

<i>Multiply</i>	<i>by</i>	<i>to obtain</i>	<i>Multiply</i>	<i>by</i>	<i>to obtain</i>
miles per hour	26.82	meters per minute	lb-force per sq in (psi)	6895	pascals (Pa)
Newtons (N)	0.2248	pound-force	slug	14.594	kilogram (kg)
Newtons (N)	0.1020	kilogram-force	square centimeters	0.1150	square inches
pascals (Pa)	0.0209	lb-force per sq ft	square centimeters	10^{-4}	square meters
pascals (Pa)	1.450×10^{-4}	lb-force per sq in (psi)	square centimeters	100	square millimeters
pascals (Pa)	0.102	kgf per sq meter	square feet	144	square inches
pascals (Pa)	0.102×10^{-4}	kgf per sq cm	square feet	0.09290	square meters
pounds	453.6	grams	square inches	6.452	square centimeters
pounds of water	0.01602	cubic feet	square inches	6.944×10^{-2}	square feet
pounds of water	27.68	cubic inches	temp (deg C) + 17.8	1.8	temp (deg F)
pounds of water per min	2.569×10^{-4}	cubic feet per sec	temp (deg F) - 32	5/9	temp (deg C)
pounds per cubic foot	16.02	kgs per cubic meter	tons (long)	2240	pounds
pounds per cubic foot	5.787×10^{-4}	pounds per cubic in	tons (long)	1016	kilograms (kg)
pounds per square foot	0.01602	feet of water	tons (short)	2000	pounds
pounds per square foot	4.882	kgs per sq meter	tons (metric)	1000	kilograms (kg)
pounds per square inch	703.1	kgs per sq meter	watts (W)	1.341×10^{-3}	horse-power
pounds per square inch	144	pounds per sq foot	watts (W)	1.360×10^{-3}	hp (metric)
pound-force	4.448	Newtons (N)	yards	0.9144	meters (m)
lb-force per sq ft	47.88	pascals (Pa)			

SI Unit Prefixes

<i>Multiplication Factor</i>	<i>Prefix</i>	<i>Symbol</i>	<i>Pronunciation (USA) (1)</i>	<i>Meaning in USA</i>	<i>In Other Countries</i>
1 000 000 000 000 000 000 = 10^{18}	exa (2)	E	ex' a (a as in about)	One quintillion times	trillion
1 000 000 000 000 000 = 10^{15}	peta (2)	P	as in <i>petal</i>	One quadrillion times	thousand billion
1 000 000 000 000 = 10^{12}	tera	T	as in <i>terrace</i>	One trillion times	billion
1 000 000 000 = 10^9	giga	G	jig' (a as in about)	One billion times	milliard
1 000 000 = 10^6	mega	M	as in <i>megaphone</i>	One million times	
1 000 = 10^3	kilo	k	as in <i>kilowatt</i>	One thousand times	
100 = 10^2	hecto	h (4)	heck'toe	One hundred times	
10 = 10^1	deka	da (4)	deck' a (a as in about)	Ten times	
0.1 = 10^{-1}	deci	d (4)	as in <i>decimal</i>	One tenth of	
0.01 = 10^{-2}	centi	c (4)	as in <i>sentiment</i>	One hundredth of	
0.001 = 10^{-3}	milli	m	as in <i>military</i>	One thousandth of	
0.000 001 = 10^{-6}	micro	μ (5)	as in <i>microphone</i>	One millionth of	
0.000 000 001 = 10^{-9}	nano	n	nan' oh (an as in ant)	One billionth of	milliardth
0.000 000 000 001 = 10^{-12}	pico	p	peek' oh	One trillionth of	billionth
0.000 000 000 000 001 = 10^{-15}	femto	f	fem' toe (<i>fem</i> as in <i>feminine</i>)	One quadrillionth of	thousand billionth
0.000 000 000 000 000 001 = 10^{-18}	atto	a	as in <i>anatomy</i>	One quintillionth of	trillionth

Glossary*

Added mass—The total hydrodynamic force, per unit acceleration, exerted on a ship or other body in phase with and proportional to the acceleration.

Advance—The distance by which the center of gravity (CG) of a ship advances in the first quadrant of a turn. It is measured parallel to the approach path, from the CG position at rudder execute to the CG position where the ship has changed heading by 90°. Maximum advance is the distance, measured parallel to the approach path from the CG position at rudder execute to the tangent to the path of the CG normal to the approach path. The first of these terms is the most commonly used.

Advance coefficient (J)—A parameter relating the speed of advance of the propeller V_A to the rate of rotation n , given by $J = V_A/nD$, where D is the propeller diameter. The advance coefficient may also be defined in terms of ship speed V , in which case it is given by $J_V = V/nD$.

Afterbody—That portion of a ship's hull abaft amidships.

After peak—The compartment in the stern, abaft the aftermost watertight bulkhead.

After perpendicular—See **length between perpendiculars**.

Amidships—In the vicinity of the midlength as distinguished from the ends. Technically it is exactly halfway between the forward and the after perpendiculars.

Amplitude—The magnitude of the extreme of a sinusoidal quantity with respect to the mean value. The double amplitude is the magnitude of the difference of the extreme.

Angle of attack (α)—The angle, measured in the plane

containing the lift vector and the inflow velocity vector, between the velocity vector representing the relative motion between a body and a fluid and a characteristic line or plane of the body, such as the chord line of an airfoil or hydrofoil. Synonymous with angle of incidence.

Appendages—The portions of a vessel extending beyond the main hull outline, including such items as rudder, shafting, struts, bossings, and bilge keels.

Archimedes' Principle—The principle that states that buoyant force acting on a body partially or completely immersed in a fluid is equal to the weight of fluid displaced.

Aspect ratio (AR)—The ratio between the span of a hydrofoil or airfoil, measured at right angles to the liquid flow, to the chord c of the hydrofoil, in the direction of flow. When the chord varies in length across the span, the aspect ratio is the span b divided by the mean chord $\bar{c}$ obtained generally by dividing the hydrofoil projected area A_p into the square of the span b , i.e. b^2/A_p .

Athwartship—Across the ship, at right angles to the fore-and-aft centerline.

Back (of blade)—The side of a propeller which faces generally in the direction of ahead motion. This side of the blade is also known as the suction side of the blade because the average pressure there is lower than the pressure on the face of the blade during normal ahead operation. This side of the blade corresponds to the upper surface of an airfoil or wing.

Ballast—Any solid or liquid weight placed in a ship to increase the draft, to change the trim, or to regulate the stability.

Baseline—A fore-and-aft reference line at the upper surface of the flat plate keel at the centerline for flush shell-plated vessels, or the thickness of the garboard strake above that level for ships having lap-

*The glossary is adapted from *Ship Design and Construction* and the *ITTC Dictionary of Ship Hydrodynamics*.

seam shell plating. Vertical dimensions are measured from a horizontal plane through the baseline, often called the molded baseline.

Beam (B)—A dimension expressing breadth or width of a body or ship in a transverse horizontal direction. When not otherwise defined the beam is the molded breadth of a ship, measured amidships at the designed waterline. According to the position where the breadth is measured, it is named
beam, extreme: maximum beam wherever it occurs on the hull above or below water.

beam, immersed, maximum: maximum beam of the underwater body.

beam, maximum section (B_X): beam measured on the designed waterline at the maximum section area.

beam, midlength (B_M): beam at the midsection of the designed waterline.

Beam, molded—The maximum breadth of the hull measured between the inboard surfaces of the side shell plating of flush-plated ships, or between the inboard surfaces of the inside strakes of lap seam-plated vessels.

Beam, deck—An athwartship horizontal structural member, usually a rolled shape, supporting a deck or flat.

Bilge—Intersection of bottom and side. May be rounded or angular as in a chine form hull. The lower parts of holds, tanks, and machinery spaces where bilge water may accumulate.

Bilge keel—A long longitudinal fin fitted at the turn of the bilge to reduce rolling. Commonly it consists of plating attached to the shell plating.

Bilge strake—Course of shell plates at the bilge.

Blockage—The effect of the boundaries of a channel or tunnel on the flow around a body.

Blockage correction—A correction made to the results of hydrodynamic experiments made in a channel or tunnel of one cross-section in order to estimate the equivalent results for another cross-section. Specifically a correction made to the results of a resistance experiment in a towing tank in order to estimate the equivalent results in unrestricted water.

Block coefficient (C_B)—The ratio of the underwater volume of a ship to the volume of a rectangular block, the dimensions of which are the effective length, draft, and beam. The relationship is expressed as a decimal.

Body plan—A drawing consisting of two half transverse elevations or end views of a ship, both having a common vertical centerline, so that the right-hand side represents the ship as seen from ahead, and the left-hand side as seen from astern. On the body plan appear the forms of the various cross sections, the curvature of the deck lines at the side, and the pro-

jections, as straight lines of the waterlines, the buttock lines, and the diagonal lines.

Bollard pull—The pull force exerted by a ship at zero ship speed. It is the sum of the propeller thrust and the interaction force on the hull.

Bonjean curves—A set of curves, each of which represents a plot of the cumulative area of a station on the lines plan, from the base line to any point above it.

Bossing or boss—The curved swelling outboard portion of the ship's shell plating that surrounds and supports the propeller shaft.

Boundary layer—The region of fluid close to a solid body where, due to viscosity, transverse gradients of velocity are large as compared with longitudinal variations, and shear stress is significant. The boundary layer may be laminar, turbulent, or transitional.

Boundary layer thickness (δ)—The distance normal to the surface of a body at which the speed attains that in an equivalent inviscid flow. For practical purposes this is sometimes taken as 99 percent of the inviscid flow speed or 98 percent of the total head.

Bow line—The intersection of the molded hull surface forward of amidships with any vertical longitudinal plane not on the centerline. See **buttock**.

Bracket—A plate used to connect rigidly two or more structural parts, such as deck beam to frame, or bulkhead stiffener to the deck or tank top (usually triangular in shape).

Breadth, molded— See **beam, molded**.

Breakwater—Inclined bulwark-like structure on a weather deck to deflect overboard water coming over the bow or over the gunwale and moving aft.

Breasthook—A triangular plate bracket joining port and starboard side stringers at the stem.

Broaching—An involuntary and dangerous change in heading produced by a severe following or quartering sea.

Bulkhead—A term applied to the vertical partition walls that subdivide the interior of a ship into compartments or rooms. The various types of bulkheads are distinguished by their location, use, kind of material, or method of fabrication, such as forepeak, longitudinal, transverse, watertight, wire mesh, and pilaster. Bulkheads that contribute to the strength of a vessel are called strength bulkheads, and those that are essential to the watertight subdivision are watertight or oiltight bulkheads. Gastight bulkheads serve to prevent the passage of gas or fumes.

Bulkhead, after peak—A term applied to the first main transverse bulkhead forward of the sternpost. This bulkhead forms the forward boundary of the after peak tank.

Bulkhead, collision or forepeak—The foremost main

- transverse watertight bulkhead. It extends from the bottom of the hold to the freeboard deck and is designed to keep water out of the forward hold in case of bow collision damage.
- Bulkhead deck**—The bulkhead deck is the uppermost deck up to which the transverse watertight bulkheads and shell are carried.
- Bulkhead, screen**—A term applied to a light nonwatertight transverse bulkhead fitted in some Great Lakes ore carriers. Its greater flexibility allows it to survive the effects of the unloading machinery.
- Buttock**—The intersection of the molded surface abaft amidships with any vertical longitudinal plane not on the centerline. See **bow line**.
- Camber**—The rise or crown of a deck, athwartship; also called round of beam.
- Camber (of a hydrofoil)**—The maximum separation of the mean line and the nose-tail line.
- Capacity plan**—A plan outlining the spaces available for cargo, fuel, fresh water, water ballast, etc., and containing cubic or weight-capacity lists for such spaces and a scale showing deadweight capacities at varying drafts and displacements.
- Casing, engine and boiler**—Bulkheads enclosing a large opening between the weather deck and the engine and boiler rooms. This provides space for the boiler uptakes and access to these rooms, and permits installing or removing large propulsion units such as boilers or turbines.
- Cathodic protection**—Protection of a ship's hull against corrosion by superimposing on the hull an impressed current, provided by a remote power source through a small number of inert anodes. Also accomplished by fitting aluminum, magnesium, or zinc anodes in the underwater portion of a ship or in tanks, which waste away by galvanic action.
- Cavitation**—In most engineering contexts, the process of formation of the vapor phase of a liquid when it is subjected to reduced pressure at constant ambient temperature. In general, a liquid is said to cavitate when vapor bubbles form and grow as a consequence of pressure reduction.
- Cavitation damage**—Deformation and/or erosion of materials in cavitated regions, associated primarily with the high pressures developed during cavity collapse.
- Celerity**—See **wave speed**.
- Center girder**—A vertical plate on the ship's centerline between the flat keel and inner bottom or rider plate, extending the length of the ship. Also called center vertical keel (CVK) or center keelson.
- Centerline** (☉)—The middle line of the ship, extending from stem to stern at any level.
- Center of buoyancy** (B)—The geometric centroid of the submerged volume of a body or ship through which the total buoyancy may be assumed to act. Its position, measured as the distance from midships or from the fore ($\overline{FB}$) or after perpendicular ($\overline{AB}$) is called the **longitudinal center of buoyancy** and from the base line or keel ($\overline{KB}$) the **vertical center of buoyancy**.
- Center of flotation** (F)—The geometric centroid of the area of the waterplane of any waterline. Its position measured as the distance from midships or from the fore or after perpendicular is called **longitudinal center of flotation** and is generally expressed as a ratio of the waterline length.
- Center of gravity** (G)—The center through which all the weights constituting the ship and its contents may be assumed to act. The distance measured from midships, the fore perpendicular ($\overline{FG}$) or the after perpendicular ($\overline{AG}$), and from the baseline or keel ($\overline{KG}$) are called **longitudinal** and **vertical center of gravity**, respectively.
- Chine**—A more or less sharp corner or knuckle in the hull form, continuous over a significant length of the ship, as in the junction of side and the bottom in planing craft. The chine is known as soft when the corner is rounded, and hard otherwise.
- Chord (of a hydrofoil, airfoil, or propeller blade)** (c)—The length of the straight line connecting the extremities of the mean line of a hydrofoil section. It passes through, or nearly through, the fore and aft extremities of the section. Synonymous with nosetail line.
- Chord length, mean** ($\bar{c}$)—The quotient obtained by dividing the expanded or developed area of a propeller blade by the span from the hub to the tip.
- Coefficient, drag** (C_D)—A relationship between the drag D of a ship or body and the dynamic pressure times a specified area. It is customary to express it as $C_D = D/qA$.
- Coefficient of lift** (C_L)—A relationship between the lift force L developed by a ship or body and the dynamic pressure times a specified area. It is customary to express it as $C_L = L/qA$.
- Cofferdam**—Narrow void space between two bulkheads or floors that prevents leakage between the adjoining compartments.
- Collision bulkhead**—See **bulkhead, collision**.
- Compartmentation**—The subdividing of the hull by transverse watertight bulkheads so that the ship may remain afloat under certain assumed conditions of flooding.
- Control surfaces**—Control surfaces are the rudders, planes, and other hinged or movable devices used for controlling the motion of a body or ship.
- Correlation allowance coefficient**—For the ship-model

correlation allowance, the resistance coefficient of the form C_A corresponding to the resistance form R_A , the suffix A is to denote the additional resistance to be added to the smooth-ship prediction to complete the ship-model balance. This allowance covers not only such items as roughness allowance but also the method of extrapolation used and the scale effects on resistance, wake, thrust deduction, and other propulsive factors. A coefficient of this kind lends itself to subdivision into different components, such as those due to structural roughness (C_{AS}), paint roughness (C_{AP}), and so on.

Correlation allowance, model-ship (R_A)—This is the addition that has to be made to the resistance of the “smooth” ship, as predicted from the model results, to bring it into agreement with the actual ship performance determined from full-scale trial or service results. The correlation allowance depends upon the method used to extrapolate the model results to the smooth ship, the ship length and type, the basic shell roughness of the newly painted ship, fouling, weather conditions at the time the ship measurements were taken, and scale effects on the factors making up the model and ship propulsive coefficients.

Counter—The overhanging portion of the stern of a ship, which lies between the designed waterline and the deck and which projects abaft the waterline termination.

Coupling—The influence of one mode of motion on another, for instance, coupling between heave and pitch.

Damping—A characteristic property of a dynamic system, which dissipates energy and reduces motion.

Damping coefficient—The ratio of damping force or moment amplitude to velocity amplitude as a function of frequency.

Dead flat—The portion of a ship’s structure that has the same transverse shape as the midship frame. See **parallel middle-body**.

Deadrise—Athwartship rise of the bottom from the keel to the bilge.

Deadweight—The carrying capacity of a ship at any draft and water density. Includes weight of cargo, fuel, lubricating oil, fresh water in tanks, stores, passengers and baggage, and crew and its effects.

Deadwood—The portion of the hull of a vessel that displaces a volume small in proportion to its weight, but which is provided for the support of a propeller shaft, structural strength, docking support, protection when grounding, or for other reasons. It is placed generally at the after end, but not necessarily on the center line. Specifically, the thin portion of the hull at the stern of a vessel immediately forward of or in the vicinity of the rudder.

Deck—A platform in a ship corresponding to a floor in

a building. It is the plating, planking, or covering of any tier of beams either in the hull or superstructure of a ship.

Deck, bulkhead—See **bulkhead deck**.

Deck, freeboard—Deck to which freeboard is measured; the uppermost continuous deck having permanent means of closing all weather openings.

Deck, platform—A lower deck, usually in the cargo space, that does not contribute to the longitudinal strength of the ship.

Deck, shelter—See **shelter deck**.

Deck, tonnage—The tonnage deck constitutes the upper boundary of the internal volume of the measurable portions of the ship as defined by the tonnage regulations.

Deck, weather—Uppermost continuous deck with no overhead protection.

Deck beam—See **beam**.

Deck height—The vertical distance between the molded lines of two adjacent decks.

Deck stringer—The strake of deck plating that runs along the outboard edge of a deck.

Deckhouse—An enclosed erection on or above the weather deck that does not extend from side to side of the ship.

Deep tanks—Tanks extending from the bottom or inner bottom up to or higher than the lowest deck. They are often fitted with hatches so that they also may be used for dry cargo in lieu of fuel oil, ballast water, or liquid cargo.

Density, mass (ρ)—The mass per unit volume of a substance.

Depth, molded—The vertical distance from the molded baseline to the tip of the freeboard deck beam at side, measured at mid-length of the ship.

Diagonal—The trace on the outside of a body marking the intersection of a plane passing through it at an angle other than 90° to the baseplane.

Diameter, steady-turning—The diameter of the circular arc described by the center of gravity of a ship when it has achieved a steady-turning state.

Diameter, tactical—The distance travelled by the center of gravity of the ship normal to its original approach path in turning through 180° . Tactical diameter is equal to the transfer at 180° change of heading.

Displacement, light—The weight of the ship including hull, machinery, outfit, equipment, and liquids in machinery.

Displacement, loaded (Δ)—The displacement of a ship when floating at her greatest allowable draft. It is equal to the weight of water displaced and is the sum of the light displacement and the deadweight.

Double bottom—Compartments at the bottom of a ship between inner bottom and the shell plating, used for ballast water, fresh water, fuel oil, etc.

Doubling (doubler) plate—A plate fitted outside or inside of and faying (touching) against another to give extra local strength or stiffness.

Draft (T)—The depth of the ship below the waterline measured vertically to the lowest part of the hull, propellers, or other reference point. When measured to the lowest projecting portion of the vessel, it is called the **extreme draft**, when measured at the bow, it is called **forward draft**, and when measured at the stern, the **after draft**. The average of the forward draft and the after draft is the **mean draft**, and the mean draft when in full-load condition is the **load draft**.

Draft marks—The numbers on each side of a ship at the bow and stern, and sometimes amidships, to indicate the distance from the lower edge of the number to the bottom of the keel or other fixed reference point. The numbers are 6 inches high and spaced 12 inches bottom to bottom vertically in English units.

Drag—The designed excess of draft aft over that forward when fore and aft drafts are measured from the designed waterline.

Drag (D)—The fluid force acting on a moving body in such a way as to oppose its motion; the component of the fluid forces parallel to the axis of motion of a body. Drag is the preferred term in aerodynamics and for submerged hydrodynamic bodies, while resistance is generally used in ship hydrodynamics. The various forms of drag are defined in relation to resistance. See also **resistance**.

Drag coefficient (C_D)—The non-dimensional ratio of the drag per unit of a representative area of a body to the dynamic pressure far ahead of the body.

Dynamic positioning—A means of holding a ship in a relatively fixed position with respect to the ocean floor without using anchors, accomplished by two or more propulsive devices controlled by inputs from sonic instruments on the sea bottom and on the ship, by gyrocompass, by satellite navigation, or by other means.

Dynamic pressure—See **pressure, dynamic**.

Effective length (L)—The length used for speed-power calculations and the coefficients for it. Effective length is determined from the sectional area curve by excluding any abrupt tailing off at the after end of the curve such as often occurs with single-screw, cruiser sternships. In multiscrew normal vessels, it is usually the load waterline length, but in single-screw ships with either cruiser or fantail sterns, it is usually the length from the forward perpendicular to about the middle of the propeller aperture.

Efficiency, hull (η_H)—The ratio between the useful work done on the ship and the work done by the propeller or other propulsion devices in a given time that is effective power P_E and thrust power P_T respectively.

Efficiency, propeller, behind hull (η_B)—The ratio between the power P_T developed by the thrust of the propeller and the power P_D absorbed by the propeller when operating behind a model or ship.

Efficiency, propulsive or quasi-propulsive (η_D)—The ratio between the useful or effective power P_E and the power delivered to the propeller or the propulsion device P_D .

Efficiency, relative rotative (η_R)—The relative rotative efficiency is the ratio of the propeller efficiencies behind the hull and in open water, as already defined.

Efficiency, shafting (η_S)—The shafting efficiency is a measure of the power lost in shaft bearings and stern tube.

Efficiency, propeller, open water (η_O)—The ratio between the power developed by the thrust of the propeller P_T and the power absorbed by the propeller P_D when operating in open water with uniform inflow velocity V_A .

Entrance—That portion of a ship's body forward of the parallel middlebody or the point at which the slope of the sectional area curve is zero.

Even keel—This term is used to define the condition in which the ship has her keel parallel to the water surface. For vessels in which the keel is not straight or normally parallel to the water surface, "zero trim" or "level trim" is preferred to even keel.

Expansion trunk or tank—A trunk extending above a space which is used for the stowage of liquid cargo. The surface of the cargo liquid is kept sufficiently high in the trunk to permit expansion without risk of excessive strain on the hull or of overflowing, and to allow contraction of the liquid without increase of free surface.

Face (of blade)—The side of the propeller blade which faces downstream during ahead motion. This side of the blade is also known as the pressure side because the average pressure on the face of the blade is higher than the average pressure on the back of the blade during normal operation. The face corresponds to the lower surface of an airfoil or wing.

Face plate—Generally a narrow stiffening plate fitted along the inner edge of web frames, stringers, etc., to form the flange of the member.

Factor, magnification—The ratio of output amplitude at a certain frequency to input amplitude.

Factor, tuning (Λ)—The ratio of excitation frequency to natural frequency (or the ratio of natural period of a motion to period of encounter).

Fair—To smooth or fair up a ship's lines; eliminating irregularities; to assemble the parts of a ship so that they will be fair, i.e., without kinks, bumps, or waves.

Fairwater—A term applied to plating fitted around the ends of shaft tubes and strut barrels, and shaped to

streamline the parts, thus eliminating abrupt changes in the waterflow. Also applied to any casting or plating fitted to the hull for the purpose of preserving a smooth flow of water.

Fantail—The overhanging stern section of ships that have round or elliptical after endings to uppermost decks and that extend well abaft the after perpendicular. Also called **counter**.

Faying surface—The surface between two adjoining parts.

Fin—A fixed or movable hydrofoil, attached to a ship generally in a longitudinal direction, to improve the dynamic stability or maneuverability, or to provide a lift force to windward, as in the fin keel of a sailing yacht.

Flange—The part of a plate or shape bent at right angles to the main part; to bend over to form an angle. See also **face plate**.

Flap—A hinged movable auxiliary hydrofoil forming the aftermost portion of a main hydrofoil.

Flare—The spreading out of the hull form from the central vertical plane, with increasing rapidity as it rises from the waterline to the rail; usually in the forebody. Also a night distress signal.

Flat—A partial deck, usually without camber or sheer.

Floodable length—The length of ship that may be flooded without sinking below her safety or margin line. The floodable length of a vessel varies from point to point throughout her length and is usually greatest amidships and least near the quarter length.

Floor—Vertical transverse plate immediately above the bottom shell plating, often located at every frame, extending from bilge to bilge.

Flow, laminar—The flow of a viscous liquid in which layers of laminae of fluid appear to slide smoothly past each other. Momentum transfer and shear between neighboring layers of fluid are due to molecular interactions only.

Flow, régime—A term referring to the state of the flow in any region; the principal recognized régimes are laminar, transitional, turbulent, and separated flows.

Flow, reversed—Flow occurring in an eddy or in a separated zone in which the local flow has a component opposite in direction to that of the main flow.

Flow, secondary—A transverse flow induced by the boundary layer geometry and by pressure conditions existing in the main flow.

Flow, separated—The detachment of the main fluid flow from a solid surface due to an adverse longitudinal pressure gradient sometimes caused by a sudden change of the direction or the curvature of the surface. The fluid in the separated flow contains eddies and may be nearly static or may contain a region of reversed flow.

Flow, steady—Flow in which the velocity pattern is independent of time.

Flow, transitional—An unstable state of viscous flow between the laminar and turbulent régimes.

Flow, turbulent—A flow in which there are rapid and apparently random fluctuations both in the magnitude and in the direction of velocity. The velocity fluctuations may also be described by a random spectrum of vortices of varying size and strength. Turbulent resistance is higher than that in laminar flow at the same Reynolds number, because of the high momentum exchange by transverse fluctuations.

Flow, uniform—Flow in which all velocity vectors are parallel and equal.

Flow, viscous—The flow of a fluid where the flow characteristics include the effects of the shear forces acting on the fluid and within it.

Fluid, perfect or ideal—A hypothetical fluid which is homogeneous, inviscid, and incompressible.

Flush deck ship—A ship constructed with an upper deck extending throughout her entire length without a break or a superstructure such as forecastle, bridge, or poop.

Force, damping—A force that tends to reduce motion and, if assumed to be linear, is proportional to the velocity.

Force, exciting—A fluctuating external force that causes motion of a body, as when a ship encounters a train of waves.

Force, restoring—A force tending to return a body to its equilibrium position when it has been displaced by an external force.

Fore—A term used to indicate the portion or portions of a ship at or adjacent to the bow. Also applied to the parts of the ship lying between amidships and the stem, as forebody, forehold, and foremast.

Forebody—That portion of the ship's body forward of amidships.

Forecastle—A superstructure fitted at the extreme forward end of the upper deck.

Forefoot—The lower end of a ship's stem, which curves to meet the keel.

Forepeak—The watertight compartment at the extreme forward end. The forward trimming tank.

Form effect—The difference between the viscous resistance of a model or a ship and the two-dimensional frictional resistance of a flat plate of the same length and wetted area and at the same speed in a given fluid. The difference arises because of the augmented speed of flow around the ship form as compared with that along a flat plate and the pressure resistance of viscous origin. See also **form factor**.

Form factor (r or k)—The ratio between the total viscous resistance coefficient of a model or a ship C_V and the two-dimensional frictional resistance coefficient of a flat plate C_{FO} at the same free stream Reynolds number.

Forward or fore perpendicular (FP)—See **length between perpendiculars**.

Foundation—The structural supports for the boilers, main engines or turbines, and reduction gears are called main foundations. Supports for machinery space auxiliary machinery are called auxiliary foundations. Deck machinery supports are called steering engine foundation, winch foundation, etc.

Frame—A term used to designate one of the transverse members that make up the rib-like part of the skeleton of a ship. The frames act as stiffeners, holding the outside plating in shape and maintaining the transverse form of the ship. See also **longitudinal**.

Frame spacing—The fore-and-aft distance, heel to heel, of adjacent transverse frames.

Freeboard—The distance from the waterline to the upper surface of the freeboard deck at side.

Freeboard deck—See **deck**, **freeboard**.

Freeing port—An opening in the lower portion of a bulwark, which allows deck water to drain overboard. Some freeing ports have hinged gates that allow water to drain overboard but that swing shut to prevent seawater flowing inboard.

Free surface—Liquid in a partially filled tank or compartment that tends to remain horizontal as the vessel heels or rolls.

Frequency (f)—The number of cycles occurring per unit of time: $f = 1/T$, where T is the period.

Frequency, circular (ω)—In any cyclic motion, or in any periodic motion that may be represented by a cyclic motion, the circular frequency is the angular velocity. If ω is in radians per second, $\omega = 2\pi/T$, where T is the period.

Froude number (F_n)—A dimensionless parameter expressing the condition of dynamical similarity for flow systems influenced by gravity and inertia alone. In particular it defines the speed at which geometrically similar models and ship will develop wave systems that are geometrically similar. It is given by: $F_n = v/\sqrt{gL}$. The length term L is usually the length of the ship. Other forms of the Froude number use some other characteristic dimension, such as the cube root of the volume of displacement, the submergence depth, or the depth of water in restricted waterways.

Full scantling ship—A ship designed with scantlings and weather-deck closing arrangements qualifying the ship for minimum freeboard, measured from the uppermost continuous deck, according to the International Load Line Convention.

Gangway—A passageway, side shell opening, or ladderway used for boarding a ship.

Garboard strake—The strake of bottom shell plating adjacent to the keel plate.

Geosim—One of a series of models that differ in absolute size but are geometrically similar. It is a con-

traction of the expression “geometrically similar model.”

Girder—A continuous member running fore and aft under a deck for the purpose of supporting the deck beams and deck. The girder is generally supported by widely spaced pillars. Also, the vertical fore-and-aft plate members on the bottom of single or double-bottom ships.

Girth—The distance around the perimeter of any transverse station, section, or frame, between two selected points. For wetted-surface calculations, these two points are generally the waterplane intersections.

Graving dock—A structure for taking a ship out of water, consisting of an excavation in the shoreline to a depth at least equal to the draft of ships to be handled, closed at the water side end by a movable gate, and provided with large capacity pumps for removing water; blocks support the ship when the dock is pumped out.

Gravitational acceleration (g)—The acceleration, due to the earth’s gravitational field, of a freely falling body in a vacuum. This is not strictly constant and over the earth’s surface it varies by as much as ½ percent. For most terrestrial engineering purposes it is usual to disregard this variation, and for convenience the following international standard value has been agreed on: 9.80665 m/s² (32.1737 ft/s²).

Green water—Water shipped on the deck of a ship in heavy seas, as distinct from spray.

Ground tackle—A general term for anchors, cables, wire ropes, etc., used in anchoring a ship to the bottom.

Group velocity—The average rate of advance of the energy of a finite train of gravity waves.

Gudgeon—Bosses or lugs on sternpost drilled for the pins (pintles) on which the rudder hinges.

Gyradius (radius of gyration) (k_{xx} , k_{yy} , k_{zz})—The square root of the ratio of mass moment of inertia (referred to body axes) to the mass of a body.

Harmonic—Sinusoidal, in referring to a function or motion.

Hatch (hatchway)—An opening in a deck through which cargo and stores are loaded or unloaded.

Hatch coaming—The vertical plating bounding a hatch for the purpose of stiffening the edges of the opening and resisting entry of water below.

Head (h)—The height of a given fluid that the pressure in question would support.

Heading—The instantaneous direction of the projection of the forward longitudinal axis of a ship in a horizontal plane, defined by points of the compass or degrees of azimuth.

Heaving—The vertical oscillatory motion of a specified point in a vessel, usually the center of gravity. Al-

- though the heaving of a ship is a motion confined to operation in waves, it is possible with a high-speed planing craft for such motions to occur in calm water under some conditions. See **porpoising**.
- Heave-to**—To maintain control of a ship, especially in extremely heavy weather, with the minimum possible speed through the water.
- Heel or list**—A steady inclination of a ship about a longitudinal axis; to be distinguished from rolling, which is an oscillatory motion.
- Hogging**—Straining of the ship that tends to make the bow and stern lower than the middle portion. See **sagging**.
- Holds**—The large spaces below deck for the stowage of cargo; the lowermost cargo compartments.
- Horn, rudder**—A heavy casting or weldment projecting down from the hull immediately abaft the propeller to support the gudgeon fitted to take the single pintle of a semi-balanced rudder.
- Hub**—The central portion of a screw propeller to which the blades are attached and through which the driving shaft is fitted. Also known as the **boss**.
- Hull**—The structural body of a ship, including shell plating, framing, decks, bulkheads, etc.
- Hull girder**—That part of the hull structural material effective in the longitudinal strength of the ship as a whole, which may be treated as analogous to a girder.
- Hydroelasticity**—The study of the interaction between the inertial, hydrodynamic, and elastic forces in a structure subjected to hydrodynamic loading. Divided into dynamic hydroelasticity, where these three forces are co-existent, or static hydroelasticity where inertial forces are absent. Analogous to aeroelasticity.
- Hydrofoil**—A structure externally similar to an airplane wing designed to produce lift and operate in water.
- Hydrofoil, span (b)**—The length of a hydrofoil from tip to tip, from root to tip if cantilevered, or from end support to end support, measured normal to the direction of relative liquid motion.
- Inboard**—Inside the ship; toward the centerline.
- Inner bottom**—Plating forming the top of the double bottom; also called tank top.
- Intercostal**—Made in separate parts: between floors, frames or beams, etc.; the opposite of continuous.
- Keel**—The principal fore-and-aft component of a ship's framing, located along the centerline of the bottom and connected to the stem and stern frames. Floors or bottom transverses are attached to the keel.
- Keel, center vertical**—The vertical centerline web of the keel.
- Keel, flat plate**—The horizontal, centerline, bottom shell strake constituting the lower flange of the keel.
- Keel blocks**—Heavy wood, steel, or concrete blocks on which the ship rests during construction.
- Keelson, side**—Fore-and-aft vertical plate member located above the bottom shell on each side of the center vertical keel and some distance therefrom.
- Kinematic viscosity**—See **viscosity, coefficient of kinematic**.
- Knee, beam**—Bracket connecting a deck beam and frame.
- Knot (Kt)**—A unit of speed, equaling one nautical mile per hour; the international nautical mile is 1,852m (6,076.1 ft).
- Knuckle**—An abrupt change in direction of the plating, frames, keel, deck, or other structure of a ship.
- Lap**—A joint in which one part overlaps the other.
- Laying off**—The development of the lines of ship's form on the mold-loft floor and the making of templates from those lines. Also called laying down.
- Left-handed propeller**—A propeller that rotates in the counterclockwise direction when viewed from astern.
- Length, effective**—See **effective length**.
- Length, mean wetted, of a planing craft**—The mean length of the portion of the bottom of a planing craft actually wetted when under way.
- Length, overall (L_{OA})**—The extreme length of a ship measured from the foremost point of the stem to the aftermost part of the stern.
- Length between perpendiculars (L_{pp})**—The length of a ship between the forward and after perpendiculars. The forward perpendicular is a vertical line at the intersection of the fore side of the stem and the summer load waterline. The after perpendicular is a vertical line at the intersection of the summer load line and the after side of the rudder post or sternpost, or the centerline of the rudder stock if there is no rudder post or sternpost.
- Lift (L)**—The fluid force acting on a body in a direction perpendicular to the motion of the body relative to the fluid.
- Lightening hole**—A hole cut in a structural member to reduce its weight.
- Lightship weight**—See **displacement, light**.
- Liner**—A flat or tapered strip placed under a plate or shape to bring it in line with another part that it overlaps; a filler. A high-speed passenger or cargo vessel.
- Lines**—A drawing depicting the form of a ship to the molded shape and dimensions showing the stations (transverse sections or frames), waterlines, bowlines, buttocks, and profile. From the lines drawn full size on the mold-loft floor, templates are made for the various parts of the hull. See also **molded lines**.
- Line shafting**—Sections of main-propulsion shafting between the machinery and the tail shaft.
- List**—If the centerline plane of a ship is not vertical, as when there is more weight on one side than on the other, she is said to list or to heel.
- Load waterline (L_{WL})**—The line on the **lines plan** of a

ship, representing the intersection of the ship's form with the plane of the water surface when the ship is floating at the **summer freeboard** draft or at the designed draft.

Lofting—The process of developing the size and shape of components of the ship from the designed lines; traditionally, making templates using full scale lines laid down on the floor of the mold loft; today, largely performed at small scale using photographic or computer methods.

Long-crested seas—A wave system in which all components advance in the same direction. Two-dimensional waves.

Longitudinals—Fore-and-aft structural shape or plate members attached to the underside of decks, flats, or to the inner bottom, or on the inboard side of the shell plating, in association with widely spaced transverse, in the longitudinal framing system.

Maneuverability—The quality that determines the ease with which the speed, attitude, and direction of motion of a body can be changed or maintained by its control devices.

Margin line—A line, not less than 3 inches below the top of the bulkhead deck at side, defining the highest permissible location on the side of the ship of any damage waterplane in the final condition of sinkage, trim, and heel.

Margin plate—The outboard strake of the inner bottom. When the margin plate is turned down at the bilge it forms the outboard boundary of the double bottom, connecting the inner bottom to the shell plating at the bilge.

Maximum section coefficient (C_x)—The ratio of the area of the maximum vertical transverse cross section of the underwater body of a ship to the product of the waterline beam and the draft at that section.

Mean line—The mean line is the locus of the midpoint between the upper and lower surface of an airfoil or hydrofoil section. The thickness is generally measured in the direction normal to the chord rather than to the mean line. The maximum distance between the mean line and the chord line, measured normal to the chord line, is called the **camber**. The term camber line is often used synonymously with mean line.

Metacenter, transverse (M) and longitudinal (M_L)—The intersection of the vertical through the center of buoyancy of an inclined body or ship with the upright vertical when the angle of inclination approaches zero as a limit, for transverse or longitudinal inclinations respectively.

Metacenter, transverse and longitudinal; height above the baseplane (KM and KM_L)—The height, measured vertically, of the transverse or longitudinal

metacenter above the baseplane of a ship in the upright position.

Metacentric height, transverse ($\overline{GM}$) and longitudinal ($\overline{GM}_L$)—The distance between the center of gravity and the transverse or longitudinal metacenter, measured vertically in the equilibrium position. It is positive when M is above G when the ship is said to have *positive metacentric stability*; i.e., on inclination to a small angle a restoring moment arises that acts to return the ship to the vertical.

Metacentric radius, transverse ($\overline{BM}$) and longitudinal ($\overline{BM}_L$)—The height, measured vertically, of the transverse or longitudinal metacenter above the center of buoyancy of a ship in the upright position. Geometrically, $\overline{BM}$ is the radius of curvature of the locus of the center of buoyancy related to transverse inclinations, and $\overline{BM}_L$ the radius of curvature of the locus of the center of buoyancy related to longitudinal inclinations.

Midship—See **amidships**.

Midship section—A drawing showing a typical cross section of the hull and superstructure (when appropriate) at or near amidships, and giving the scantlings of the principal structural members.

Molded—An adjective used to indicate the generally fair form and dimensions of the hull as determined by the lines to the inside of the shell plating.

Molded lines—Lines defining the geometry of a hull as a surface without thickness; structural members are related to molded lines according to standard practice (unless otherwise shown on drawings), e.g., the inside surface of flush shell plating is on the molded line, also the underside of deck plating.

Mold loft—A floor space for laying down (laying off) the full-size lines of a ship and for making templates to lay out the hull's structural components.

Moment, exciting—A fluctuating external moment that causes motion of a body or ship when encountering a train of waves.

Moment, pitching—Exciting moment in pitch.

Moment of area, second (or moment of inertia)—The summation of the products of the elements of an area or surface and the squares of their distances from a given axis, generally in the surface. Specifically for a ship:

second moment of the waterplane area (or moment of inertia), longitudinal (I_L) about the transverse axis through the center of flotation.

second moment of the waterplane area (or moment of inertia), transverse (I_T) about the longitudinal axis through the center of flotation, generally the intersection of the waterplane and the centerplane.

second moment of free-water surface (or moment of inertia) generally within a ship, calculated about an

axis passing through the center of area of that surface, parallel to the expected heeling or rolling axis.

Moments of inertia (I_{xx} , I_{yy} , I_{zz}) (mass)—The summation of products of elementary masses and the squares of their distances from the respective body axes through the center of gravity—equal to the mass times the square of the **gyradius** or radius of gyration.

Mooring—Securing a ship at a pier or elsewhere by several lines or cables so as to limit her movement.

Natural period of motions: heave, pitch, roll (T_z , T_θ , T_ϕ)—The time for one complete cycle of the motion resulting when a body is displaced in calm water from its equilibrium position by an external force and then is released.

Nautical mile—A distance of 1,852 m (6,076.1 ft). See **knot**.

OBO—Abbreviation for a vessel designed to carry oil, bulk cargos, or ore cargos.

Offset—One of a series of distances, measured from reference planes (normally from the centerplane), used for defining the size and shape of a body or ship.

One-compartment subdivision—A standard of subdivision of a ship by bulkheads, which will result in the ship remaining afloat with any one compartment flooded, under specified conditions as to permeability of the compartment and the draft of the ship before flooding of the compartment.

Outboard—Abreast or away from the centerline towards the side; outside the hull.

Overhang—Any portion of the above-water hull of a ship that, when projected downward on to the designed waterplane, lies outside that designed waterline; it may be at the bow or stern, or anywhere along the sides.

Panel line—A production line where individual plates, framing members, webs, etc., are successively welded together to form an assembly unit that may include some items of outfit.

Panting—The pulsation in and out of the bow and stern plating as the ship alternately rises and plunges deep into the water. May also occur abreast the propellers of a multiscrew ship.

Parallel middlebody—The amidship portion of a ship within which the contour of the underwater hull form is unchanged.

Period (T)—The length of time for one complete cycle of a periodic quantity or phenomenon, such as the rolling of a ship from port to starboard and back to port.

Period of encounter of a ship in waves (T_e)—The time of a periodic quantity or phenomenon, such as the rolling of a ship from port to starboard and back to port.

Perpendiculars—Straight lines perpendicular to the designed load waterline of a ship through a fixed point as stated by classification rules. Specifically:

aft or after perpendicular (AP): through a fixed point at the stern; generally the aft side of the stern post, or centerline of the rudder stock in ships without a stern post.

forward perpendicular (FP): through a fixed point at the bow; generally the intersection of the fore side of the stem with the load waterline.

Pillar—Vertical member or column giving support to a deck girder, flat, or similar structure. Also called **stanchion**.

Pintles—The pins or bolts that hinge the rudder to the gudgeons on the sternpost or rudder post.

Pitch (P)—The pitch of a propeller blade section at radius r is given by $P = 2\pi r \tan \phi$ where ϕ is the angle between the intersection of the chord line of the section and a plane normal to the propeller axis. This angle is called the pitch angle. Also called **geometric pitch**.

Pitch, variable—A propeller blade for which the pitch is not the same at all radii is said to have variable pitch or varied pitch. A propeller which has the same pitch at all radii is said to be a constant pitch propeller.

Pitching—The angular component of the oscillatory motion of a hull about a transverse axis. Although pitching of a ship is a motion confined to operation in waves, it is possible with a high-speed planing craft for such motions to occur in calm water under some conditions. See **porpoising**.

Pitch ratio—The ratio of the pitch to the diameter of the propeller. Generally, the face pitch or geometric pitch at the 70-percent radius is used to compute the pitch ratio. Any measure of pitch can be used with the diameter to form a pitch ratio.

Poop—A superstructure fitted at the after end of the upper deck.

Porpoising—The oscillation of a high-speed craft, primarily in calm water, in which heaving motion is combined with pitching motion. The motion is sustained by energy drawn from the thrust.

Pounding—The impact of a water surface against the side or bottom of a ship hull, whether caused by ship velocity, water velocity, or both. Pounding is differentiated from slamming in that the impact, although heavy, is not in the nature of a shock.

Power, brake (P_B)—The power measured at the engine coupling by means of mechanical, hydraulic, or electrical brake.

Power, effective (P_E)—The power required to tow a ship, usually without her propulsive device, at constant speed v in unlimited undisturbed water: $P_E = R_T v$. The power may be for the ship either

with or without appendages. If the latter, it is usually known as the naked or bare hull, effective power.

Power, shaft (P_s)—The power delivered to the shafting system by the propelling machinery.

Power, thrust (P_T)—The power developed by the propeller thrust T at the speed of advance V_A :
 $P_T = TV_A$.

Power delivered (P_D)—The power delivered to the propeller: $P_D = 2\pi Qn$.

Power in waves, mean increase in (P_{AW})—The mean increase in power in wind and waves as compared with the power in still water at the same mean speed.

Pressure, dynamic (q)—The pressure change corresponding to the reduction of the momentum of a fluid element to zero, $q = \frac{1}{2}\rho v^2$.

Pressure, stagnation—The total pressure measured at a stagnation point.

Pressure, static—The static pressure p at a point in a stream flow is that which would be recorded by a pressure gauge advancing with the speed of the local fluid and thus static with respect to it.

Pressure, total—This is the sum of the static and dynamic pressures.

Prismatic coefficient (C_p)—The ratio of the volume of displacement to the volume of the cylinder having the length L and cross section of the maximum section of the ship. This is sometimes called the longitudinal prismatic coefficient and is given by: $C_p = \nabla/LA_x$. The prismatic coefficient can also be referred to the different parts of the ship, such as afterbody, forebody, entrance, and run. In any case the assumed length, as well as the cross section area if different from the above, is to be clearly indicated.

Prismatic coefficient, vertical (C_{VP})—The ratio of the volume of displacement to the volume of a vertical cylinder having as horizontal section the waterline and as height the draught at midships. It is given by: $C_{VP} = \nabla/A_w T$. When different, the draft of the transverse section having maximum area is used (T_x).

Profile—The outline of a ship when projected on the fore-and-aft vertical plane; also, the outline of parts of the ship, such as the stem, stern, and rudder, when similarly projected.

Note: This definition also covers the contour of any flat or curved surface that acts as a hydrofoil or as a control surface; examples are the profiles of diving planes on submarines, fitted generally in a horizontal plane, and the profiles of the blades on a screw propeller.

Propeller—Most generally, any device that will produce thrust to propel a vehicle. The most common form is the screw propeller, which basically consists of a central hub and a number of fixed blades ex-

tending out radially from the hub. Lift is generated by the blades when the propeller is rotated. One component of the lift force produces the desired thrust and the other component creates torque, which must be overcome by the engine to sustain rotation.

basic screw propeller: a propeller that may be described as fixed pitch, subcavitating, open (unducted), and fully submerged. Variations on this basic type are listed below.

adjustable-pitch propeller: a propeller whose blades can be adjusted to different pitch settings when the propeller is stopped.

contrarotating propeller: two propellers rotating in opposite directions on coaxial shafts.

controllable-pitch propeller: a propeller having blades that can be rotated about a radial axis so as to change the effective pitch of the blades while the propeller is operating. This allows full power to be absorbed for all loading conditions. If the pitch can be adjusted to the extent that reverse thrust can be achieved without reversing the direction of rotation of the shaft then the propeller is sometimes called a controllable-reversible-pitch propeller.

cycloidal propeller: a propeller consisting of a flat disc set flush with the under surface of the vessel with a number of vertical, rudder-like blades projecting from it. The disc revolves about a central axis and each of the blades rotates about its own vertical axis. The axis of each blade traces a cycloidal path. The blade motion can be varied so as to produce a net thrust in any desired direction in a plane normal to the axis of rotation. It is used where excellent maneuverability is required.

ducted propeller: a propeller with a short duct mounted concentrically with the shaft. The duct, or nozzle, is shaped so as to control the expansion or contraction of the slipstream in the immediate vicinity of the propeller. In one form (the Kort nozzle) the flow is accelerated, whereas in the other form (pump jet) the flow is decelerated. A pump jet is sometimes also defined as a ducted propeller with stator vanes regardless of whether the flow is accelerated or decelerated.

fully cavitating propeller: a propeller designed to operate efficiently at very low cavitation numbers where a fully developed cavity extends at least to the trailing edge of the blade. The blade sections of such propellers have relatively sharp leading edges for more efficient supercavitating operation and thick trailing edges for strength. Also known as *supercavitating propeller*.

interface propeller: a propeller of the fully cavitating ventilated type designed to operate with only a portion of the full disc area immersed. These propellers are considered for high speed applications to vehicles

such as surface effect ships where the appendage drag associated with the shafts and struts of a fully submerged propeller would result in a considerable increase in resistance. Also known as partially submerged or surface propellers.

ring propeller: a propeller with a very short duct attached to the tips of the blades and rotating with the propeller. Also called a banded propeller.

steerable ducted propeller: a ducted propeller in which the duct can be pivoted about a vertical axis so as to obtain a steering effect.

supercavitating propeller: see *fully cavitating propeller*.

tandem propeller: two propellers fitted to the same shaft, one behind the other, and rotating as one.

ventilated propeller: a propeller of the fully cavitating type, but with provision to introduce air into the cavities in order to achieve fully developed, stable cavities at lower speeds than would otherwise be possible.

vertical axis propeller: synonymous with *cycloidal propeller*.

Quenching—In steelmaking, an operation consisting of heating the material to a certain temperature and holding it at that temperature to obtain desired crystalline structure, and then rapidly cooling it in a suitable medium such as water or oil. Quenching is often followed by tempering.

Rabbit—A groove, depression, or offset in a member into which the end or edge of another member is fitted, generally so that the two surfaces are flush. A rabbit in the stem or stern frame would take the ends or edges of the shell plating, resulting in a flush surface.

Rail—The rounded member at the upper edge of the bulwark, or the horizontal pipes or chains forming a fencelike railing fitted instead of a bulwark.

Rake—A term applied to the fore-and-aft inclination from the vertical, of the mast, smokestack, stem, etc. In river and some ocean barges, it is the end portion of the hull, in which the bottom rises from the midship portion to meet the deck at the headlog.

Resistance (R)—The fluid force acting on a moving body in such a way as to oppose its motion; the component of the fluid forces acting parallel to the axis of motion of a body. Resistance is the preferred term in ship hydrodynamics, while drag is generally used in aerodynamics and for submerged bodies. Total resistance is denoted by R_T . See also **drag**.

Resistance, coefficient (C_F , C_R , C_S , C_T , C_V , C_W , etc.)—The nondimensional ratio of any specific component of resistance per unit area to the dynamic pressure far ahead of the body.

Resistance, frictional (R_F)—The component of resist-

ance obtained by integrating the tangential stresses over the surface of a body in the direction of motion.

Resistance, frictional specific (C_F)—An alternative name for the coefficient of frictional resistance, in which the reference area is taken to be the wetted area under consideration.

Resistance, pressure (R_P)—The component of resistance obtained by integrating the normal stresses over the surface of a body in the direction of motion.

Resistance, residuary (R_R)—A quantity obtained by subtracting from the total resistance of a hull a calculated friction resistance obtained by any specific formulation.

Resistance, spray (R_S)—The component of resistance associated with the expenditure of energy in generating spray.

Resistance, viscous (R_V)—The component of resistance associated with the expenditure of energy in viscous effects.

Resistance, viscous pressure (R_{PV})—The component of resistance obtained by integrating the components of the normal stresses due to viscosity and turbulence. This quantity cannot be directly measured except for a fully submerged body when it is equal to the pressure resistance R_P .

Resistance, wavebreaking (R_{WB})—A resistance component associated with the breakdown of the ship bow wave.

Resistance, wavemaking (R_W)—The component of resistance associated with the expenditure of energy in generating gravity waves.

Resistance, wave pattern (R_{WP})—A resistance component deduced from measurements of wave elevations remote from the ship or model where it is assumed that the subsurface velocity field, and hence the momentum of the fluid, can be related to the wave pattern by means of linearized theory. The resistance so deduced does not include wavebreaking resistance.

Resistance in waves, mean increase in (R_{AW})—The mean increase in resistance in wind and waves as compared with the still-water resistance at the same speed.

Resonance—The dynamical condition of a simple, uncoupled system where the excitation frequency is equal to the natural frequency.

Note: In a coupled system, the dynamic condition where the excitation frequency corresponds to the frequency of maximum response to unit exciting force over a range of frequencies.

Response amplitude operator—The square of the ratio of response amplitude to excitation amplitude of a forced harmonic motion applied to a linear system as a function of frequency.

Revolutions per minute, mean increase in waves—The mean absolute increase in revolutions per minute, as

compared with those in calm water, necessary to maintain speed in wind and waves.

Reynolds number (R_n)—A dimensionless parameter expressing the condition of dynamical similarity for flow systems influenced by viscosity and inertia alone. For equal values of Reynolds number and the same orientation to the flow, the specific resistance coefficients of all geometrically similar smooth surfaces are identical as long as the uninfluenced speed fields are similar and the flow is influenced by viscosity and inertia alone. It is given by: $R_n = \nu L \rho / \mu = \nu L / \nu$. The length term L is usually the length of the surface, but the distance from the leading edge of the surface to a specific point, the diameter of a body, or the thickness of the boundary layer is sometimes used as a length term.

Rider plate—A continuous flat plate attached to the top or bottom of a girder.

Rigging—Chains, wire ropes, fiber lines, and associated fittings and accessories used to support masts and booms used for handling cargo and stores and for other purposes.

Rise of floor—See **deadrise**.

Right-handed propeller—A propeller that rotates in the clockwise direction when viewed from astern.

Righting arm—The horizontal distance between the center of gravity and the vertical line of action through the center of buoyancy at any angle of heel.

Righting moment—The product of the righting arm and the displacement of a ship at any angle of heel.

Rolling—The angular component of the oscillatory motion of a ship measured about a longitudinal axis.

Root—The part of the propeller blade adjacent to the propeller hub.

Ro-Ro or Ro/Ro—Abbreviation for a vessel designed to carry vehicles, so arranged that the vehicles may be loaded and unloaded by being rolled on or off on their own and/or auxiliary wheels, via ramps fitted in the sides, bow, or stern of the vessel.

Rudder—A control surface which by its action or movement controls the steering or the turning of a ship in a horizontal plane. Specifically, a hinged or movable control-surface appendage in the form of a hydrofoil, placed either at the bow or at the stern of a ship, or at both ends, to apply a turning moment to the ship. Rudder types are listed below.

balanced or semi-balanced: a control surface in the form of a swinging rudder in which a fraction of the area is placed forward of the vertical turning axis to reduce the operating torque in the ahead direction.
compound: a control device in the form of a fixed vertical appendage, to the after edge of which is hinged a movable or swinging rudder. See also *flap*.

contra: a rudder with a curved blade, designed to be

mounted abaft a propeller to take advantage of the rotation in the slipstream and to produce a forward thrust on the rudder.

flap: a control device in the form of a moving rudder that is hinged for practically its entire vertical height to the hull, to a skeg, or to a fin which has an area large in proportion to that of the rudder. This type of rudder takes its name from the flaps on airplane wings; both function by building up large pressure differentials on the fixed parts of the ship or airplane to which they are attached.

offset: a rudder offset from the centerplane of a ship either to port or starboard.

spade: a control device in the form of a moving appendage that projects below the stern of the ship without any fixed supports in front of it or below it.

Rudder post—A vertical or nearly vertical member of the ship's structure upon which the steering rudder is hung or supported.

Rudder stock—The portion of the rudder, concentric with the axis of rotation, which provides bearing support and also transmits the operating torque.

Rudder stop—Lug on stern frame or a stout bracket on deck at each side of the quadrant, to limit the swing of the rudder to approximately 37 degrees port or starboard. A rudder angle of 35 degrees is the maximum usually used at sea (45 degrees on inland waterway vessels).

Run—That part of a ship's body aft of the parallel middle body or the point at which the slope of the sectional area curve is zero.

Sagging—Straining of the ship that tends to make the middle portion lower than the bow and stern. See **hogging**.

Scale effect—The change in any force, moment, or pressure coefficients, flow pattern, or the like, due to a change in absolute size between geometrically similar models, bodies, or ships. These variations in performance due to differences in absolute size arise from the inability to satisfy simultaneously all the relevant laws of dynamical similarity (e.g., gravitational, viscous, and surface tension).

Scantlings—The dimensions of a ship's frames, girders, plating, etc.

Scantling draft—The maximum draft at which a vessel complies with the governing strength requirements. Usually used when the scantling draft is less than the geometrical draft corresponding to the freeboard calculated according to the load line convention.

Scarf—A connection made between two pieces by tapering their ends so that they fit together in a joint of the same breadth and depth as the pieces connected. It is used on bar keels, stem and stern frames, and other parts.

Scoop—An opening in the surface of the underwater

body of a ship, which may or may not be fitted with a projection extending beyond that surface, designed for catching and taking water into a ship.

Scuppers—Drains from decks to carry off accumulations of rainwater, condensation, or seawater. Scuppers are located in the gutters or waterways, on open decks, and connect to pipes usually leading overboard and in corners of enclosed decks to the bilge.

Sea chest—An enclosure, attached to the inside of the underwater shell and open to the sea, fitted with a portable strainer plate. A sea valve and piping connected to the sea chest passes seawater into the ship for cooling, fire, or sanitary purposes. Compressed air or steam connections may be provided to remove ice or other obstructions.

Seakeeping—In general, a term covering the study of the behavior and performance of ships in a seaway. As an adjective, a term signifying a ship's ability to maintain normal functions at sea.

Seakindliness—The quality of behaving comfortably in a seaway; that property of a ship producing easy motions in a seaway.

Seam—Fore-and-aft joint of shell plating, deck and tank top plating, or a lengthwise edge joint of any plating.

Section—The intersection of a plane with a body or ship that it passes through in any position or direction; specifically for a ship, any transverse section perpendicular to the designed waterplane, such as the 1) area, maximum section (A_X) or 2) area, midship section, midlength section, midsection, or midstation section (A_M).

Sectional area coefficients (C_X), (C_M)—The *maximum transverse section coefficient* C_X is given by $C_X = A_X/B_X T_X$ where A_X is the area of the maximum transverse section; B_X and T_X are the beam and draft at this section respectively. The *midship section coefficient* C_M is given by $C_M = A_M/B_M T$ where A_M is the midship section area; B_M and T are the beam and draft at midships respectively.

Sectional area curve—A diagram of transverse section areas up to the designed waterline plotted on a base of length L , representing the distribution of underwater volume along the length of a ship; this diagram may be made dimensionless by plotting each ordinate as the ratio of the area A of any section to the area A_X of the maximum section and by plotting the position of that section as a fraction of the ship length L along the base from selected reference points (generally forward and after perpendiculars or midships).

Sections—A general term referring to structural bars, rolled or extruded in any cross section, such as angles, channels, bulbs, Tees, H- and I-bars (or beams). Sometimes called **profiles**.

Shaft tunnel, shaft alley—A watertight enclosure for the propeller shafting large enough to walk in, extending

aft from the engine room to provide access and protection to the shafting in way of holds.

Shape—A rolled bar of constant cross section such as an angle, bulb angle, channel, etc; also to impart curvature to a plate or other member.

Shear stress (τ)—The tangential resisting force per unit area acting on a boundary. In a viscous fluid, the shear stress is the tangential resisting force per unit area acting on any boundary within the fluid. The specific value of the shear stress at a wall is denoted by τ_w .

Sheer—The longitudinal curve of a vessel's decks in a vertical plane, the usual reference being to the ship's side; in the case of a deck having a camber, its centerline sheer may also be given in offsets. Due to sheer, a vessel's deck height above the baseline is higher at the ends than amidships.

Sheer strake—The course of shell plating at strength deck level.

Shell plating—The plates forming the outer side and bottom skin of the hull.

Ship—A vessel intended for marine transportation, without regard to form, rig, or means of propulsion.

Short-crested seas—An irregular wave system in which the components advance in various directions. Three-dimensional waves.

Shoulder—The portion of a ship at the junction of the middlebody with the entrance or the run, where the waterlines approach or reach their maximum width.

Singing—Intense discrete frequency sound radiated from the propeller when the blades vibrate. Generally thought to be due to the shedding of Karman vortices from the trailing edges of the blades at a resonant frequency of the blade vibration.

Sinkage—The steady-state lowering of a ship's position of flotation in the water; to be distinguished from heaving, which is an oscillatory motion.

SI units—The system of units now being used internationally is the Systeme International d'Unites (SI). The conversion table in appendix C provides conversion factors for U.S. Customary units, and MKS units that differ from SI units, which are used in this volume. Proper use of significant figures and rounding-off techniques should be given due consideration when using the conversion factors.

Skeg—A projection from a fixed appendage applied to the underwater hull of a ship, generally to increase the lateral area and give increased swing damping and dynamic stability to the hull. A skeg is usually of large lateral area compared to its transverse thickness, is usually fitted in a vertical plane, and is in the after part of the vessel.

Skew-back—The displacement of any blade section along the pitch helix measured from the generator line to the reference point of the section. Positive skew-back

- is opposite to the direction of ahead motion of the blade section. Also called skew.
- Slamming**—Heavy impact resulting from a vessel's bottom forward making sudden contact with the sea surface after having risen on a wave. Similar action results from rapid immersion of the bow in vessels with large flare.
- Span (b)**—The distance from tip to tip of a hydrofoil or airfoil. The distance from root to tip in the semi-span.
- Specific gravity**—The dimensionless ratio of the weight of unit volume of the designated substance to the weight of unit volume of fresh water.
- Spectral density, one dimensional ($S(\omega)$)**—A function of frequency whose integral over any interval represents the energy contribution of all the component waves of a random function in that interval.
- Speed, corresponding**—The speed of a ship v_s related to that of a model v_M , or vice-versa, according to Froude's law of comparison: $v_s = v_M \sqrt{\lambda}$, where λ is the scale factor.
- Speed, ground**—The speed of a ship relative to the ground, that is the speed including the effects of tide and currents. When the ship is moving through still water the ground speed is the same as the true water speed.
- Speed, hump (in high-speed craft)**—The speed at which the resistance reaches a maximum before a planing craft enters the planing phase, or a hydrofoil craft enters the foilborne phase.
- Speed of advance of a propeller (v_A)**—When a propeller behind a ship or model is producing the same thrust at the same rate of rotation as in open water the corresponding speed v_A determined from the open-water propeller characteristics is termed the speed of advance of the propeller. This is usually less than the ship speed v_s .
- Springing**—A vibration of the complete vessel induced by wave forces in conjunction with the ship's elastic properties. More pronounced in ships having a high length-to-depth ratio.
- Squatting**—The increase in trim by the stern assumed by a ship when under way over that existing when at rest.
- Stability**—The tendency of a ship to remain upright or the ability to return to her normal upright position when heeled by the action of waves, wind, etc.
- Stanchion**—Vertical column supporting decks, flats, girders, etc; also called a **pillar**. Rail stanchions are vertical metal columns on which fence-like rails are mounted. See **rail**.
- Steering gear**—A term applied to the steering wheels, leads, steering engine, and fittings by which the rudder is turned. Usually applied to the steering engine.
- Stem**—The bow frame forming the apex of the intersection of the forward sides of a ship. It is rigidly connected at its lower end to the keel and may be a heavy flat bar or of rounded plate construction.
- Stern, cruiser**—A spoon-shaped stern used on most merchant ships, designed to give maximum immersed length.
- Stern, transom**—A square-ended stern used to provide additional hull volume and deck space aft and(or) to decrease resistance in some high-speed ships.
- Stern frame**—Large casting, forging, or weldment attached to the after end of the keel. Incorporates the rudder gudgeons and in single-screw ships includes the rudder post.
- Sternpost**—Sometimes, the vertical part of the stern frame to which the rudder is attached.
- Stern tube**—The watertight tube enclosing and supporting the tail-shaft. It consists of a cast-iron or cast-steel cylinder fitted with bearing surface within which the tailshaft, enclosed in a sleeve, rotates.
- Stiff**—A vessel is said to be stiff if she has an abnormally large metacentric height. Such a ship may have a short period of roll and therefore will roll uncomfortably. The opposite of **tender**.
- Stiffener**—An angle, T-bar, channel, built-up section, etc., used to stiffen plating of a bulkhead, etc.
- Stiffness**—A relative term indicating resistance to deformation or bending.
- Strain**—The deformation resulting from a stress, measured by the ratio of the change to the total value of the dimension in which the change occurred.
- Strake**—A course, or row, of shell, deck, bulkhead, or other plating.
- Streamline**—A line in a fluid such that its tangent at any point is parallel to the instantaneous velocity of the fluid at that point.
- Strength deck**—The deck that is designed as the uppermost part of the main-hull longitudinal strength girder. The bottom shell plating forms the lowermost part of this girder.
- Stress**—The force per unit section area producing deformation in a body.
- Stringer**—A term applied to a fore-and-aft girder running along the side of a ship at the shell and also to the outboard strake of plating on any deck.
- Stringer bar**—The angle connecting the deck plating to the shell plating or to the inside of the frames. The strength deck stringer bar is usually called the gunwale bar.
- Strut**—Outboard column-like support or V-arranged supports for the propeller shaft, used on some ships with more than one propeller instead of bossings.
- Suction side**—The low-pressure side of a propeller blade. Synonymous with the back of the propeller blade. Analogous to the upper surface of a wing.

Supercavitating flows—Flows in which attached, fully developed cavities extend beyond the trailing edge of the body about which the cavity is formed.

Superstructure—A decked-over structure above the upper deck, the outboard sides of which are formed by the shell plating as distinguished from a deckhouse that does not extend outboard to the ship's sides.

Surface, wetted (S)—The surface area of the underwater body of a ship. This generally includes the area of the appendages that give an appreciable contribution to the frictional drag, such as bilge keels, propeller bossings, and rudder.

Surging—The longitudinal oscillatory motion of a specified point in a ship, usually the center of gravity (or origin of body axes).

Swash bulkhead, swash plate—Longitudinal or transverse nontight bulkhead fitted in a tank to decrease the swashing action of the liquid contents as a ship rolls and pitches at sea. Their function is greatest when the tanks are partially filled. Without them the unrestricted action of the liquid against the sides of the tank might be severe. A plate serving this purpose but not extending to the bottom of the tank is called a swash plate.

Swaying—The transverse oscillatory motion of a specified point in the ship, usually the center of gravity.

Tailshaft—The aftermost section of the propulsion shafting in the stern tube in single-screw ships and in the struts of multiple-screw ships, to which the propeller is fitted.

Tank, trimming—A tank located near the ends of a ship. Seawater (or fuel oil) is carried in such tanks as necessary to change trim.

Tank, wing—Tanks located well outboard adjacent to the side shell plating, often consisting of a continuation of the double bottom up the sides to a deck or flat.

Telegraph—An apparatus, either electrical or mechanical, for transmitting orders, as from a ship's bridge to the engine room, steering gear room, or elsewhere about the ship.

Tempering—After quenching, the material is reheated to a predetermined temperature below the critical range and then cooled. In steelmaking this is done to relieve stresses set up by quenching and to restore ductility.

Template—Wood or paper full-size patterns to be placed on materials to indicate the size and location of holes, plate edges, etc; also to indicate the curvature to which frames, for example, are to be bent.

Tender—A vessel is said to be tender if she has an abnormally small metacentric height. Such a ship may have a long period of roll but may list excessively in a strong wind and may be dangerous if a hold is

flooded following a collision. The opposite of stiff. Also a small general utility boat carried aboard ship.

Thrust (T)—The force developed by a screw propeller in the direction of the shaft.

Tiller—An arm, attached to rudder stock, which turns the rudder.

Tonnage, gross—Under vessel-measurement rules of various nations, a measure of the internal volume of spaces within a vessel in which one ton is equivalent to 2.83 m³ or 100 cu ft. Under the International Convention on Tonnage Measurement of Ships (ICTM), 1969, a standardized numerical value that is a logarithmic function of spaces within a vessel. There is no definition of a ton under ICTM because the value per unit of volume is greater on a vessel of large volume than on a vessel of small volume. Gross tonnage according to the national and canal rules generally includes spaces bounded by the under surface of the uppermost complete deck, the side frames, and the floor frames or the inner bottom if it rests on the floors or if the double bottom is for water ballast, plus closed-in space in deck structures available for cargo or stores or for the berthing or accommodation of passengers or crew. Rules vary greatly as to exclusion or inclusion of various spaces. Gross tonnage according to ICTM is $GT = K_1 V$ in which V is the total molded volume of all enclosed spaces of the ship in m³ and K_1 is $0.2 + 0.02 \log_{10} V$.

Tonnage, net—Net tonnage, according to national and canal rules, is derived from gross tonnage by deducting an allowance for the propelling machinery space and certain other spaces. Net tonnage according to ICTM is a logarithmic function of the volume of cargo space, the draft to-depth ratio, the number of passengers to be accommodated, and the gross tonnage.

Torque (Q)—The torque delivered to the propeller aft of all bearings.

Tramp ship—A general breakbulk cargo ship that has no set trade route or schedule.

Transformation temperature—The temperature above which the ferrite form of iron in shipbuilding and other steel is transformed to the austenite form, and below which the ferrite form recurs. The microstructure of the steel is changed upon passing through this temperature.

Transverse—A deep member supporting longitudinal frames of bottom or side shell or longitudinal deck beams. At right angles to the fore-and-aft centerline.

Trim—The difference between the draft forward and the draft aft. If the draft forward is the greater, the vessel is said to "trim by the head." If the draft aft is the greater, she is "trimming by the stern." To trim a ship is to adjust the location of cargo, fuel,

etc., so as to result in the desired drafts forward and aft.

Tumblehome—The slant inward from the vertical of a transverse section of a hull above the designed waterline. It is the opposite of flare.

Vapor pressure—The pressure of vapor in equilibrium with its liquid state. It is then also called the saturated vapour pressure or vapour tension, which for a given substance depends only upon the temperature.

Viscosity, coefficient of dynamic (μ)—The ratio of the shearing stress in a fluid to its rate of shear deformation.

Viscosity, coefficient of kinematic (ν)—The ratio of the coefficient of dynamic viscosity to the mass density of fluid.

Wake—The wake is a term used to describe the motion imparted to the water by the passage of the ship's hull. It is considered to be positive if its direction is the same as that of the ship.

Water, restricted—A term describing a body of water in which the boundaries are close enough to the ship to affect her resistance, speed, attitude, maneuvering, and other performance characteristics, as compared with her corresponding characteristics in an open, unlimited, body of water. Principally, "restricted" applies to the proximity of the water boundaries in a horizontal direction.

Water, shallow—A term describing a body of water in which the boundaries are close enough to the ship in a vertical direction to affect her resistance, speed, attitude, maneuvering, or other performance characteristics as compared with her corresponding characteristics in water of unlimited depth.

Water, standard fresh—Water having zero salinity and a temperature of 15° C (59° F), with density $\rho = 999.00 \text{ kg/m}^3$ (1.9384 lbs/ft³), kinematic viscosity $\nu = 1.13902 \times 10^{-6} \text{ m}^2/\text{s}$ (1.22603 $\times 10^{-5}$ ft²/s).

Water, standard salt—Water having 3.5 percent salinity and a temperature of 15° C (59° F), with density $\rho = 1,025.87 \text{ kg/m}^3$ (1.9905 lbs/ft³), kinematic viscosity $\nu = 1.18831 \times 10^{-6} \text{ m}^2/\text{s}$ (1.27908 $\times 10^{-5}$ ft²/s).

Waterline—The intersection line of the free-water surface with the molded surface of a ship, either in still water or when she is surrounded by waves of her own making. The intersection line of any selected plane, parallel to the baseplane, with the molded surface of a ship. The angle of the waterline at the bow in the horizontal plane neglecting local shape at stem is the angle of entrance. The angle of the waterline at the stern in the horizontal plane neglecting local shape of stern frame is the angle of run.

Waterplane—Any selected plane through a ship form and parallel to the baseplane, specifically
designed waterplane: corresponding to the designed waterline.

maximum waterplane: corresponding to the waterline of a ship at the draft at which the waterplane area is maximum.

Waterplane area (A_w)—The area enclosed by a waterline.

Wave—A disturbance of the surface of a fluid that usually progresses across the surface as the result of circular or other local motions of the fluid components. A standing wave is a special case of a wave that does not advance.

amplitude (ζ_A): the radius of orbital motion of a surface wave particle, equal to one half of the wave height.

components: the waves of different frequencies and directions found by spectral analysis to compose an irregular sea, or the large number of finite waves used to approximate such an irregular sea.

direction, angle of (μ): the angle between the direction of a component wave and the x_0 axis.

encounter, angle of (μ_E): the angle between the longitudinal axis of the ship and the direction of the wave encounter.

frequency (f): the reciprocal of wave period = $1/T$; or circular frequency (ω) = $2\pi/T$.

height (ζ_w , H): the vertical distance from wave crest to wave trough, or twice the wave amplitude of a harmonic wave.

height, apparent ($\tilde{\zeta}_w$, $\tilde{H}$): the vertical distance between a successive crest and trough.

height, significant: the average apparent height of the third highest waves in an irregular pattern.

instantaneous elevation (ζ): the instantaneous elevation of a point in a wave system above the level of the undisturbed surface.

length (L_w , λ): the horizontal distance between adjacent wave crests in the direction of advance.

length, apparent ($\tilde{L}_w$, $\tilde{\lambda}$): the horizontal distance between adjacent wave crests of an irregular sea in the direction of advance.

$$\text{number } (\kappa): \kappa = \frac{2\pi}{\lambda} \text{ or } \frac{2\pi}{L_w}$$

period (T): time between the passage of two successive wave crests past a fixed point.

period, apparent ($\tilde{T}$): the time elapsing between the occurrence of two successive crests of an irregular sea, or between two successive upward crossings of zero in a record.

profile: the elevation of the surface particles of a wave plotted as a function of space in fixed time.

slope of surface: the surface slope of a wave profile perpendicular to the crest in space coordinates. Maximum wave slope of a regular harmonic or trochoidal wave is $\pi/H/L_w$.

speed or celerity (c_w): the phase velocity of a surface gravity wave in deep water.

$$c_w = \sqrt{\frac{gL_w}{2\pi}}$$

steepness ratio: the ratio of wave height to length.

train: a continuous sequence of wave crests and hollows.

trochoidal: a profile closely approximating that of a regular surface gravity wave in a fluid; it can be geometrically constructed by tracing the path of a point on the radius of a circle as the circle rolls along the underside of a horizontal line.

Web frame—A built-up frame to provide extra strength, usually consisting of a web plate flanged or otherwise stiffened on its edge, spaced several frame spaces apart, with the smaller, regular frames in between.

Wetness—The quality of a part of the ship, usually the weatherdeck forward, with respect to its liability of being wet as a result of motions of ship and waves.

Whipping—The transient ship-hull vibration induced by impulsive excitation forces. For example, fore-bottom slamming, bow-flare slamming, shipping of water and stern slamming. See also **springing**.

Yawing—The angular component of the oscillatory motion of a hull about a vertical axis.

References

- Beer, F. P.; and Johnston, E. R. 1977. *Vector mechanics for engineers: statics*. 3rd ed. New York: McGraw-Hill.
- Bhattacharyya, R. 1978. *Dynamics of marine vehicles*. New York: John Wiley.
- Bissel, A. M.; Oertel, E. J.; and Livingston, D. J. 1976. *Shipboard damage control*. Annapolis: Naval Institute Press.
- Breyer, S. 1973. *Battleships and battle cruisers, 1905–1970*. Garden City: Doubleday.
- Brown, D. 1977. The introduction of the screw propeller into the Royal Navy. *Warship* 1, no. 1:59–63. Annapolis: Naval Institute Press.
- Brunel, I. 1972. *The life of Isambard Kingdom Brunel*. Cranbury, England: Associated University Presses.
- Cohen, S. 1981. *Proceedings of the 19th American Towing Tank Conference*. 2 vols. Ann Arbor: Ann Arbor Science Publishers.
- Cokelet, E. D. 1978. Breaking waves: the plunging jet and interior flowfield. *Proceedings of the symposium on mechanics of wave-induced forces on cylinders*. San Francisco: Pitman.
- Comstock, J. P., ed. 1967. *Principles of naval architecture*. New York: SNAME.
- Corlett, E. 1974. *The iron ship*. New York: Arco Publishing.
- Cox, G. G.; and Lloyd, A. R. 1977. Hydrodynamic design basis for navy ship roll motion stabilization. *Transactions of SNAME* 85:51–93.
- Dulin, R. O.; and Garzke, W. H. 1976. *Battleships: United States battleships in World War II*. Annapolis: Naval Institute Press.
- Froude, W. 1955. *The papers of William Froude*. Edited by A. D. Duckworth. London: Institution of Naval Architects.
- Gamow, G. 1961. *Biography of physics*. New York: Harper Torchbooks.
- Gertler, M. 1954. *A reanalysis of the original test data for the Taylor Standard Series*. TMB report 806. Bethesda: DTNSRDC.
- Gillmer, T. C. 1975. *Modern ship design*. 2d ed. Annapolis: Naval Institute Press.
- Hibbeler, R. C. 1978. *Engineering mechanics: statics*. 2d ed. New York: Macmillan.
- Higdon, A.; Ohlsen, E. H.; Stiles, W. B.; Weese, J. A.; and Riler, W. F. 1976. *Mechanics of materials*. 3rd ed. New York: John Wiley.
- Hoyt, J. W. 1972. The effect of additives on fluid friction. *Transactions of ASME: journal of basic engineering* 94D: 258–85.
- Inui, T. 1962. Wave-making resistance of ships. *Transactions of SNAME* 70:283–353.
- Johnson, B.; and Nehrling, B., eds. 1977. *Proceedings of the 18th American Towing Tank Conference*. 3 vols. Annapolis: U.S. Naval Academy.
- Kinsman, B. 1965. *Wind waves*. Englewood Cliffs: Prentice-Hall.
- Kracht, A. M. 1978. Design of bulbous bows. *Transactions of SNAME* 86:197–217.
- McCormick, M., 1973. *Ocean engineering wave mechanics*. New York: John Wiley.
- Moore, J. E. 1980. *Jane's fighting ships, 1980–81*. New York: Franklin Watts.
- National Research Council of Canada. 1975. *Proceedings of the 14th International Towing Tank Conference*. Ottawa: National Research Council.
- Naval Sea Systems Command. 1981. *Ship work breakdown structure*. NAVSEA Publication 0900–LP–029–9010. Washington: Naval Sea Systems Command.
- Naval Sea Systems Command. 1968. *Theory of strandings and the use of beach gear*. NAVSEA Publication 0994–000–3010. Washington: Naval Sea Systems Command.
- Nelson, L. W.; and McCallum, D. 1978. Fins of the future FFG 7. *Naval engineers journal* 90, no. 5:29–39.
- Norrby, R. A.; and Ridley, D. E. 1980. Notes on thrusters for ship maneuvering and dynamic positioning. *Transactions of SNAME* 88:377–402.
- Oosterveld, M. W. C., ed. 1978. *Proceedings of the 15th International Towing Tank Conference*. Wageningen, the Netherlands: Netherlands Ship Model Basin.
- Popov, E. P. 1968. *Introduction to mechanics of solids*. Englewood Cliffs: Prentice-Hall.

- Rawson, K. J.; and Tupper, E. C. 1976. *Basic ship theory*. 2d ed. 2 vols. New York: Longman.
- Reynolds, O. 1883. An experimental investigation of the circumstances which determine whether the motion of water shall be direct or sinuous and of the law of resistance in parallel channels. *Philosophical transactions of the Royal Society of London* 174:935–82.
- Salvesen, N.; Tuck, E. O.; and Faltinsen, O. 1970. Ship motions and sea loads. *Transactions of SNAME* 78:250–87.
- Sarchin, T. H.; and Goldberg, L. L. 1962. Stability and buoyancy criteria for U.S. naval surface vessels. *Transactions of SNAME* 70:418–58.
- Taggart, R., ed. 1980. *Ship design and construction*. New York: SNAME.

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